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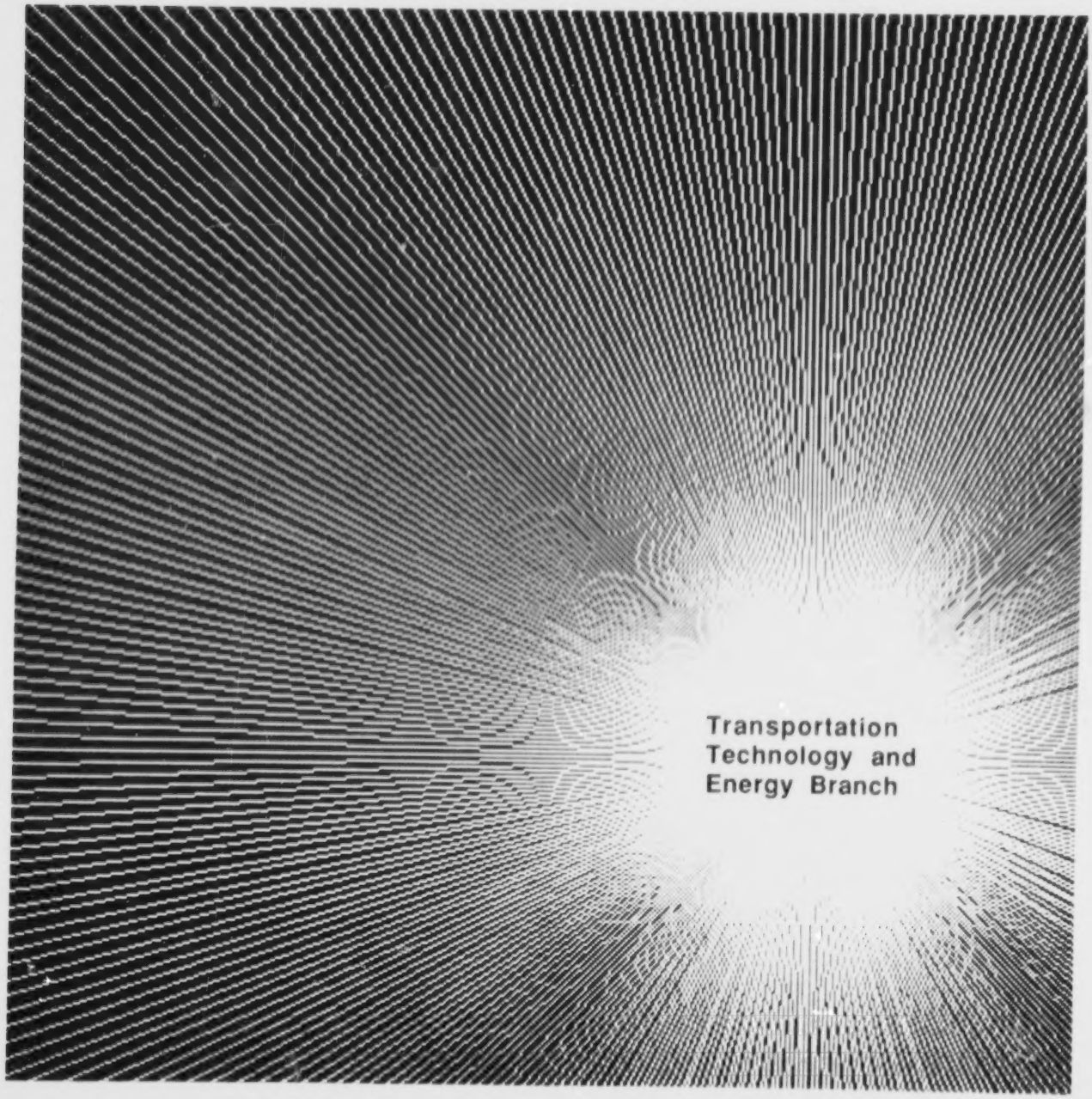

Bi-level III Ride Quality Improvement Program Final Report –

Volume I - Simulation Report
Volume II - Test Report

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Bi-level III Ride Quality Improvement Program Final Report –
Volume I - Simulation Report

Prepared by:
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Prepared for:
Transportation Technology and Energy Branch

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March 1990

Technical Report Documentation Page

Bi-level III Ride Quality Improvement Program
Final Report – Volume I - Simulation Report

Prepared by:	**UTDC Inc.**
Prepared for:	Rail and Advanced Transportation Systems Office Transportation Technology and Energy Branch Ministry of Transportation T. AppaRao, Project Manager
Report No.:	RATSO-90-01
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Abstract:	The objective of the program was to improve the vertical ride quality of the GO Transit Bi-level passenger car upper deck. A series of nine vertical dynamic models were developed to represent the car and possible low-cost methods to improve the ride. Each model was run to predict the natural frequencies and modes, and the response to random vertical and cross-level track impact. As well, a 1/4 car ANSYS-based finite element model was derived and used to assess the effect upon model frequencies and relative acceleration (pseudo-acceleration) of the various methods proposed to improve the vertical ride quality.
Key words:	commuter rail, passenger car, ride quality, vehicle dynamic simulation
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NOMENCLATURE

Symbol	Description	
Arabic		
a_b	Body vertical acceleration	m/sec^2
ca	Absorber damping	NS/m
c_{cr}	Critical damping	NS/m
F_b	Vertical body force	N
g	Acceleration due to gravity	m/sec^2
K_c	Bushing Assy Stiffness	
m_a	Absorber mass	kg
m_b	Body mass	kg
rx, ry, r3	Rotation in x, y, z direction	
tx, ty, t2	Translation in x, y or z direction	
Greek		
μ	Mass ratio = m_a/m_b	
ζ	Optimum absorber damping ratio = C_a/C_{cr}	

EXECUTIVE SUMMARY

The objective of this program was to improve the ride quality of the GO Transit Bi-level passenger car upper deck.

The upper deck was found to be subject to vertical vibration (12-16 Hz) known as "dither" during operation at high speeds. The major constraints imposed on the program were a) to improve the vertical ride quality of the upper deck such that it met a mutually acceptable standard (i.e., 2.5 hours I.S.O. reduced comfort boundary), and b) any modification devised had to be less than $10K on a production basis.

A series of nine vertical dynamic models were developed to represent the bi-level car and proposed modifications to improve the ride quality. No one modification studied was capable of reducing the dither vibration to the proposed target level.

A 1/4 car ANSYS-based finite element model was derived and used to assess the effect upon modal frequencies and relative acceleration (pseudo-accleration) of the various methods under consideration to improve the vertical ride quality. From this analysis it was determined that two modes primarily affect the dither vibration; a 12 Hz body bending mode causing both decks to vibrate equally and 15-16 Hz mode associated with a heave motion of the central portion of the upper deck.

TITLE: BI LEVEL III RIDE QUALITY IMPROVEMENT PROGRAM FINAL REPORT VOLUME I SIMULATION REPORT

1.0 INTRODUCTION

1.1 BACKGROUND

The Bi-level III Ride Improvement Program was initiated in May 1987 after completion of negotiations between MTC/UTDC and GO Transit (1). It had as its purpose the elimination of the unpleasant upper deck vertical vibration known as "dither" which exists on the GO Bi-level commuter rail car. This vibration problem has been investigated at various times by teams from Hawker Siddeley (2,3,4,5,6) (now Can Car Works, Thunder Bay), Dofasco and MTO. In 1987 March, discussions amongst MTO, GO and UTDC resulted in a two-phase program being agreed to which would address the dither problem. Phase I of this program would see two proposed fixes: a) replacement of the upper deck floor with strengthened panels and b) application of vertical primary dampers between the axle boxes and truck frame. If no acceptable solution was found in Phase I, then a more in depth analysis effort would be considered in a Phase II which would be negotiated separately. Each proposed modification to the car was to be evaluated analytically and then in a static shop test. If judged suitable, then the mod would be road tested. The chief constraint on the program was that the car modification cost on a production basis had to lie within 10K$. The program was completed in February 1989 and consisted of analytical as well as test portions. This report describes the analytical activities which were carried out.

The data available at the start of the program (2,3,4,5,6) suggested that dither was centred on the upper deck and occurred over a frequency range from 10 - 15 Hz. There was also evidence that the truck frame primary bounce frequency and wheelset rotational frequencies were in the same range and thus could excite the dither vibration. These early tests investigated the effect of wheel unbalance, drag link position, airbag stiffness and damping and shock absorber configuration. In the case of the latter, it was discovered that the dither vibration could be reduced by removing the shock absorbers or reducing shock absorber mounting bushing stiffness. However, concerns over effects upon carbody roll displacement and fatigue life of shock absorber bushings quieted any further work.

1.2 OBJECTIVES

The objectives of the program were:

a) to improve the vertical ride of the car, particularly the upper deck such that it met a mutually acceptable standard (say, 2.5 hours I.S.O. reduced comfort boundary); and

b) to ensure that the cost in production of fitting any modification package to a revenue car would be less then 10 K$;

c) to improve the roll characteristic of the car when negotiating turn-outs and cross-overs (to be done once dither problem resolved).

1.3 METHOD OF APPROACH

Because of the need to keep costs down, a more intuitive approach was adopted in which it was decided to evaluate two modifications which appeared to offer the best chance for a solution. To evaluate these analytically two distinct sets of software tools were employed.

1.3.1 Finite Element Model

The bi-level car was modelled using ANSYS several years ago at CCW, Thunder Bay. For this project, the model was updated in November of 1987 by James Ma at CCW to include the latest version of the structure. The model was exercised in the dynamic mode to predict the normal (natural) frequencies and mode shapes in order to compare modified versions of the car with the unmodified car. A report summarizing the analysis work is contained in (7). A description of the ANSYS software can be had by consulting any of the standard user's manuals.

A newly acquired in-house finite element package called PAL2 was also used in Kingston for design of the vibration absorber in Phase I. This package is essentially similar to ANSYS.

1.3.2 Dynamic Model

To assess the vertical dynamics of the bi-level car including random track inputs and thus simulate motion along tangent track, a UTDC

developed software package was used. This modelled the car with a multi-degree of freedom lumped mass configuration. Three steps were required to perform a simulation.

1. The model input file, describing the car, was run with program AGEM to generate the equations of motion.

2. The output file from AGEM was then run in the eigenvalue routine, RIDE, to obtain the natural frequencies and mode shapes then,

3. The output file from AGEM was run in the forced response routine, RPSD, to obtain the transfer function of each lumped mass and the power spectral density and one third octave spectrum of selected locations on the carbody. The runs were made for a particular speed and track class (vertical profile roughness) setting.

Typically, the upper deck centre, lower deck centre and intermediate deck over one of the trucks were chosen as standard for calculation of the output spectra. By plotting the results using program GPLOT, comparison could be made of ride spectra, acceleration spectra, etc., measured during road tests.

2.0 INPUT DATA SOURCES

This section describes the input data used to build the dynamic model of the bi-level car. Since access to any previous modelling work was not possible, a fresh approach was taken using UTDC in-house software tools. It is recognized that a model is really only as good as the assumptions used to derive it and the quality of the input data. Thus it was important to obtain accurate parameter values with which to start the process. The approach used was to build the simplest model using the data in (8) and Appendix A. It is not known to what extent this data represented measured values. However, in the end that was less important a question as the model was matched to actual test data, where possible. The track input was taken from a Battelle report (9) and used in analytical representations of vertical profile and cross-level roughness for various F.R.A. track classes.

3.0 TEST RESULTS USED TO TUNE DYNAMIC MODEL

3.1 INITIAL BASELINE

Before the evaluation of mods could occur, a proper working model of the baseline bi-level car had to be derived. The suspension parameters specified originally by Dofasco in (8) and Appendix A were used to provide the initial values used in the dynamic analysis. In order to tune the additional features of the model including the upper deck effective mass and damping, use was made of data from several tests. The first test used was the MTO ride test of 1987 April, in which acceleration narrow band and third octave spectra were produced from measurements on 18 cars (see Appendix B). This also includes 1 minute worst average third octave spectra and upper to lower deck transfer functions for six of the test cars. The relationship amongst intermediate deck, lower and upper deck was explored via Figure 38 from (5), an MTO test of bi-level car 2150 in 1984 November.

The measured resonance frequencies of the production truck frame and bolster were used to tune the model and came from Dofasco testing in the early 1980's (see Appendix C).

The above data as a set was used to tune the initial baseline version of the bi-level car model, that is, the 36 degree-of-freedom, 19 mass model stored as BILEVEL3 on the HP1000-A900 system which is reported in (10).

3.2 IMPROVED DEFINITION REFINED MODEL

The baseline model was used for theoretical assessment of the primary damper, vibration absorber (BILEVEL5) and secondary damper bush mods (BILEVEL6), all of which were done prior to the test series termed "improved definition of problem". The "improved definition" tests were carried out in 1988 June to better understand the contribution of the truck to the generation of dither vibration. From the extensive amount of data collected, a revised vertical model of the car was derived. This data has been grouped into Appendix D, Improved Definition test data, and was used to develop the models BILEVEL8 and 9 which represent the refined versions of BILEVEL3.

4.0 DYNAMIC MODEL RESULTS

To evaluate the truck primary dampers, vibration absorber, structural stiffening and secondary damper modification a vertical dynamic model of the bi-level car was constructed at UTDC using in-house software. The result is a series of models which represent various configurations of interest. The development history is given in table 4.0-1.

TABLE 4.0-1

No.	File or Model	Description
1.	BILEVEL1	35 degree of freedom (dof) model including single rigid mass for carbody plus 17 masses representing 2 trucks and eight ground masses (wheel elements).
2.	BILEVEL2	Same as BILEVEL1 except carbody represented by lump mass of 34,469 kg and lump mass of 1073 kg modelling upper deck portion involved in dither.
3.	BILEVEL3	Same as BILEVEL2 except large carbody mass divided into two equal masses of 17,771 kg plus upper deck moving mass of 1073 kg. Known as "3 mass" model and used as initial baseline.
4.	BILEVEL4	Same as BILEVEL3 except Upper Deck Mass given 4 degrees of freedom.
5.	BILEVEL5	Same as BILEVEL3 except vibration absorber modelled as tuned mass-spring-damper at centre of lower body mass.
6.	BILEVEL6	Same as BILEVEL3 except secondary damper bushing mod incorporated.

TABLE 4.0-1 (Cont'd)

No.	File or Model	Description
7.	BILEVEL7	First iteration of BILEVEL3 used to refine truck representation.
8.	BILEVEL8	Second iteration of BILEVEL3 in which carbody modelled as two masses (17,771 kg each) joined by angular spring at car c.g. plus upper deck moving mass (1073 kg.). Ultimately used as Improved Definition Refined Baseline.
9.	BILEVEL9	Third iteration of BILEVEL3 in which carbody modelled with two masses (17,771 kg. each) running the length of the car and joined at car c.g. to provide "sessions" action. Additional upper deck moving mass (1073 kg) also present.

The detailed development of each of the above models is available from the working files. A summary of the development will be given in the following sections with the discussion grouping one or more models within a particular topic.

4.1 BASELINE CARS

As it was important to model the baseline car in sufficient detail yet keep the analysis simple, the simulation was made in the frequency domain using linear modelling techniques. R.M.S. response was determined at selected points of interest and expressed in third octave format in order to match measured ride data. The model was constructed using the in house AGEM program, and, solved on the ride analysis program, RPSD. The eigenvalues were calculated separately using another program called RIDE. The input rail forcing function originated from a track geometry model at Battelle Labs, (9) and was in two forms: a) vertical profile and b) crosslevel. Data supplied by Dofasco (8) was used to size inertia, stiffness and damping terms.

The process employed here was to build the baseline model Bi-level I of the car and evaluate the vertical response of the upper deck and lower deck to heave and cross level inputs and then to "tune" the model to match as nearly as possible the measured vertical response of one car by adjusting the modal frequencies and damping. Once satisfactory agreement was achieved, the various modifications were added and the performance calculated for each one.

4.1.1 Description of Baseline Models

The vertical dynamics model for the Bi-level car was initially BILEVEL1, a 35 degree of freedom, 18 lump mass model shown in Figure 1. The model (later BILEVEL2 and 3) included the dither motion in a simple manner with an additional mass element equivalent to one quarter of the upper deck mass which was given a single degree of freedom connection to the carbody. This allowed direct comparison with measured upper deck response and assessment of the effect of truck changes. Every effort was made to use as few degrees of freedom as possible, however, the nature of the truck lateral connection to the carbody and phasing of input forces required that lateral, heave, longitudinal, roll and pitch d.o.f.s. be assigned to certain masses. Eight masses representing the rails were needed by the computer program RPSD. They were forced directly by the track input function and thus did not directly contribute to car response. Figure 2 shows the details of suspension and link connections idealized by all the models. The snubbers were not modelled and neither were the bolster longitudinal links, as no yaw degrees of freedom were required. Later refinement of the 35 d.o.f. model (BILEVEL3) required adding a nineteenth mass and 36th d.o.f. in Figure 3, representing body bending effects. This mass was inserted between the main body mass (No. 5) and dither mass (No. 18). All case studies were made using the latter version of the model.

4.1.2 Criteria for Assessment

To determine if the truck changes were worthwhile or not a suitable criterion had to be chosen for evaluation. In the case of the dither problem, model vibration data and ride measurements in ISO 2631 format were available (Appendix B). In view of the need to obtain as direct an evaluation upon rider comfort as possible, it

was decided to use vertical acceleration displayed against frequency calculated at the centre of the lower and upper decks. This is the most representative location as the dither vibration amplitudes are largest in this area. Response was calculated over a frequency range of 50 Hz and integrated into a third octave r.m.s. response.

The measured ride data had a variation from car to car and thus, without knowing specific car characteristics in advance, the response was tuned to fit one particular car (#2030). The process of tuning the model is described below.

4.1.3 Validation of Model

Since the objective of the analysis activity was to obtain a feel for the effect of modifying the car suspension or structure, validation of the model or tuning as referred to previously, was restricted to measured data available, at the centre of the car. The BILEVEL2 model was used at this point as BILEVEL1 did not model the upper deck separately from the main carbody mass.

The initial run on the model BILEVEL1 was made using the original Dofasco (8) data and setting the spring of the upper deck mass to give 13.4 Hz bounce frequency which was obtained from acceleration spectra typical of figure 4 for Car 2114 (Appendix B). The effective damping was taken as the lowest value (9%) from five cars for which transfer functions were measured between upper and lower deck motion. A typical function is shown in Figure 5 from Car 2030. The damping factor ζ, was derived from the relation:

$$R_{max} = Q = \frac{1}{2\zeta}$$

where: R_{max}, Q = Peak transfer function magnitude at resonance.

In spite of the resonance point appearing at about 16 Hz in Figure 5 it was decided to use 13.4 Hz since the ride response measurements showed more energy in that band and since other structural modes, not included in the model, could be shifting the peak energy downwards.

Another feature requiring adjustment in the initial model run was the coupled truck frame/bolster heave mode. With an initial modal frequency of 8 Hz the mode was corrected to 10.1 Hz to agree more closely with Dofasco truck test results. This was done by increasing the vertical static stiffness of the primary chevrons by 89%. At the same time the sidebearer static stiffness levels were raised by 15%. At this point the eigenvalues were obtained as shown in Table I.

TABLE 4.1-1

Calculated Modes/Freqs

Ev#	Freq - Hz	Damping %	Description
1	45.22	8.77	Bolster LC Roll in Ph Fwd/Rear
17	24.75	16.72	Bolster Heave In Ph Fwd/Rear
19	24.72	16.73	Bolster Heave Out of Ph Fwd/Rear
25	6.57	82.20	Bolster LC Roll In Ph Fwd/Rear
27	6.99	79.12	Bolster LC Roll Out of Ph Fwd/Rear
29	16.45	18.57	Frame LC Roll Out of Ph Fwd/Rear
31*	16.28	18.52	Frame LC Roll In Ph Fwd/Rear
33	16.52	11.56	Frame UC Roll Out of Ph Fwd/Rear
35	16.51	11.53	Frame UC Roll Out of Ph Fwd/Rear
37*	13.55	9.51	Upper Deck Heave
39*	10.04	22.41	Frame/Bolster Heave Out of Ph Fwd/Rear
41*	10.08	21.81	Frame/Bolster Heave In Ph Fwd/Rear
43	0.50	13.17	Body LC Roll
45	1.80	15.50	Body Pitch
47	1.05	29.03	Body UC Roll
49*	1.45	12.49	Body Heave
51	17.44	6.37	Fwd Frame Pitch
53	17.44	6.37	Rear Frame Pitch

* Produces significant upper deck vertical motion.

Note: LC = Lower Centre
UC = Upper Centre

The next stage in the validation process was to calculate and adjust the response by varying car parameters to match measured peak response at the dither (13.4 Hz) frequency and body pitch and heave frequencies (which appear at 1.6 Hz on the plots). The peak measured responses from MTC ride data (Appendix B) on Cars 2030, 2063 and 2124 were analysed to obtain the mean ratios between various peaks A, B and C whose location is shown in Figure 6 and tabulated below:

TABLE 4.1-2

Response Peak Ratios

Car No.	Lower Deck B:A	Upper Deck C:A	Dither C:B
2030	4.66	9.46	2.03
2063	1.34	3.20	2.39
2124	4.56	6.39	1.40
Mean	3.52	6.34	1.94

The initial response was found for an empty car on Class V track at 30 m/s (67.5 mph) and it showed a high body response relative to dither response.

The calculated peaks did not agree with the ratios in Table 4.1-2 and after further tuning effort, it was decided to try to match the ratios for Car 2030, the worst performer and also the most likely to be tested, rather than an "averaged" car.

The next adjustment was to raise the operating speed in the model to 80 mph (35.8 m/s) which slightly improved the ratio of dither to body response. Then, a range of secondary damping values from 17% to 80% critical was tried. The results showed a small but unacceptable improvement in the ratio of dither peak to body peak response for damping valves up to 26%. For the remainder of the runs the secondary damping was set to a value of 26%.

The body response was thus still unacceptably high relative to the dither response and it was thought that greater separation of body vertical heave and pitch modes would lower body response. Several

runs were made to examine this effect by varying pitch inertia but no significant change occurred when the pitch inertia was varied by ±10%. This is probably because pitch is not important to vertical response at the centre of gravity of the car, close to the location of the upper deck centre.

As a final effort with BILEVEL2 to improve agreement with measured data, the track class was set to class 6 and the truck chevron natural damping was reduced from 5% to 2.5%. The results are shown in Figures 6 and 7. It can be seen that in both Figures, the body response is in the correct range but that the dither responses are low compared with Figure 8. Even with a decrease to 2.5% critical damping in the chevrons, the overall dither response at 13.4 Hz did not increase significantly. A review at this stage of Figure 9 from (5) indicated the presence of amplification of car response in the centre of the car on both decks relative to the truck centre location at 10-16 Hz. This was interpreted as a manifestation of the body bending modes. Work by CCW (H. Pool) in 87 July (11) supported the fact that a bending mode was likely responsible for a peak of fairly constant strength at 10.8 Hz when a bare carshell was excited on the upper and lower decks separately. This meant that a portion of the "dither" response at 12.5 Hz was because of body structural bending and that due account of this should be taken in the model. At this stage, it was decided to change the model to include the body bending mode. To do this most simply, another rigid body called the bending mass (see Figure 11) with a single vertical d.o.f. was incorporated between the upper deck mass and main body mass. The goal was to close the gap in response between upper deck and lower deck and to lower the rigid body response at 1.6 Hz relative to the upper and lower deck response peaks at 12.5 Hz. This modification to the BILEVEL2 model resulted in BILEVEL3.

Nine parameters were available for adjusting the total response including the three body masses, and, spring and damping terms associated with each body. The best compromise was to keep the upper deck at its original value (1073 kg) and split the remaining mass evenly between the bending mass (17771 kg) and body mass (17771 kg). The bending mass natural frequency was set to 10.8 Hz and the upper deck frequency to 13.4 Hz. This model configuration improved the ratio of upper deck response to lower deck (bending mass) response but did not raise the level of each response bending mass damping to 0.05% critical and raising the upper deck damping to

25% critical. Next the sharpness of the rigid body response at 1.6 Hz was reduced by lowering the body heave frequency. This required softening the static value of secondary spring stiffness by 31%. The overall response level was correctly attained when the calculation was run for Class 6 track. The resulting response termed the baseline or "no dampers" response is shown in Figure 10. Compared to Figure 8, Figure 10 shows a slightly higher overall response and the ratios compare as below:

TABLE 4.1-3

Case	Lower Deck B:A	Upper Deck C:A	Dither C:B
Measured Car 2030	4.66	9.46	2.03
Calculated	4.14	9.23	2.26

Qualitatively the calculated results show a dip in the response at 2.5 Hz which is not evident in the measured data and which could be the result of another body mode at 2-4 Hz which is ignored by the model. In spite of the model only permitting discrete jumps in the strength of the track input it was possible to bring the response to good agreement, thus permitting the objectives of the analysis to be satisfied.

4.2 PRIMARY DAMPER MOD ON BASELINE CAR

Using the baseline model BILEVEL3, case studies were run to examine the effects of primary vertical dampers (PRIVERT on the plots) and type of track input.

To assess the dampers, a range of damper rates was used to run a parametric survey equivalent to 15% to 60% critical damping ratio based upon truck frame mass. For this survey, model BILEVEL3 was updated with appropriate primary damping values for each vertical damping element between axle and truck frame at a total of eight locations. This range encloses the working range of the proposed Houdaille Model RC damper. Figures 11 to 13 show the resulting response. Overall, the improvement to either upper deck or lower deck (Bending Mass) response is not significant as seen in Figure 14. A closer check shows that increasing damper strength

beyond 30% critical actually degrades performance. For the case considered here, at the optimum damping ratio of 26%, the ride improvement is 5 min. ISO on the upper deck and 0.2 hr. ISO on the lower deck.

The above results were generated for a pure heave input. For a pure cross-level input, the baseline car response with no dampers is shown in Figure 15. The cross-level input results in a very low response (100 times lower than the heave response at the car centre and thus can be ignored for the case of vertical ride response at the centre of the car. These results were presented in (10) and were not re-examined until the model was updated with the results of the Improved Definition of Problem test. Recalculation of the effect of the primary dampers was made with the refined baseline model in Section 9.6.

In summary, from the analysis, the ride on the upper and lower decks at the centre of the car is not expected to be significantly improved by adding truck primary dampers.

4.3 VIBRATION ABSORBER MOD ON BASELINE CAR

4.3.1 Simulation Results

The vibration absorber was proposed as a means of reducing the dither vibration without requiring structural modification of the carbody. It was proposed on the basis that the dither vibration is predominantly single frequency (12 - 12.5 Hz) and occurs when the first vertical bending made of the body is excited as opposed to the more local upper deck made at 15 - 16 Hz. Moreover, it was assumed that the first body bending mode has a node over each truck and has maximum deflection at the car centre.

A rough sizing of the absorber was carried out by assuming lump mass located at the centre of the lower deck is sprung in the vertical direction. If the moving mass of the carbody in first bending is the 17,771 kg. as in model BILEVEL3, and the acceleration level is 0.07 g at 12 Hz from data in Appendix A, then the mass required would be:

$$M_a = \frac{F_b}{g}$$

where: $m_a = \frac{m_b \times a_b}{g} = \frac{17771 \times 0.07\ g}{g} = 1{,}244$ kg.

If only half the vibration is eliminated then the absorber mass is also halved to 622 kg.

From (12), if $\mu = m_a/m_b$, the mass ratio, then the optimum amount of damping for an ideal auxiliary mass system is ζ_{opt}:

$$\zeta_{opt} = \sqrt{\frac{\mu}{2(1+\mu)}}$$

which for $\mu = \frac{622}{17771} = 0.035$

$$\zeta_{opt} = 0.13$$

If the required spring rate is K_a:

$$K_a = m_a (2\pi f)^2 = 622 (2\pi \times 12)^2$$

$$= 3.54 \text{ MN/m } (20{,}330 \text{ 16/in.})$$

and thus optimum damping C_a:

$$C_a = 2\zeta_{opt}\sqrt{K_a m_a} = 12{,}190 \text{ Ns/m } (69.7 \text{ lb sec/in.}).$$

We chose to size the absorber to reduce the dither vibration by half in order not to add more weight to the car than necessary yet to enable the requirement for the minimum ride level of 2.5 hours ISO RCB to be met.

The dynamic analysis model was next exercised to explore the effects upon lower and upper deck response of various combinations of absorber mass and damping which were tuned to either 10.9 or 12 Hz.

Two mass settings 622 kg and 1244 kg were used along with damping values equivalent to 5% and 18% critical. The key idea in modelling the response of the car to a wide frequency range of track input was to see how sensitive the ride was to the optimum ideal absorber design parameters. Table 4.3-1 gives a selection of calculated response amplitudes.

TABLE 4.3-1

Predicted Car Response to Vibration Absorber

Case	Mass (m_a) kg	Absorber Vert Freq Hz (20)	Absorber Vert Damping c/c_c (20)	Vertical Acceleration - g r.m.s./frequency		
				Absorber (20)	Upper Deck (18)	Lower Deck (19)
Base-line	0	-	-		1.2778-1/12-5	5.645.2/2.5
1	622	10.875	18%	1.745-2/15.7	1.045-1/12.5	4.62E-2/12.5
2	622	10.875	5%	1.81E-1/9.92	1.08E-1/12.5	4.78E-2/12.5
3	622	12.8	5%	2.435E-1/8.5	0.757E-1/12.5	3.424E2/12.5
4	622	12.8	18%	1.25E-1/12.5	0.972E-1/12.5	4.327E2/12.5
5	1244	12.8	18%	1.045E1/12.5	0.813E-1/12.5	3.646E2/12.5
6	1244	12.8	5%	1.819E1/12.5	0.674E-1/12.5	3.099E2/12.5
7	1244	12.8	2%	2.199E1/12.5	0.706E-1/12.5	3.267E2/12.5

Notes: 1. ISO Ride found from 0.63 x $a_{12.5\ Hz}$

The results are plotted in Figure 16 which shows that the ride is at 12.5 Hz 1/3 octave band sensitive to tuning frequency but not very much to absorber mass above 600 kg. The damping ratio which is best for this band is obviously 5% or even less. However, a low value will tend to focus the absorber to a narrow frequency range. Since, we are dealing with a relatively narrow range, it was decided to design the absorber with zero damping in order to assess its effectiveness in the 12 Hz area, then to add damping if the measured performance warranted.

4.3.2 Design of Absorber

Having estimated the absorber mass, spring and damping characteristics in 4.3.1, the next task was to design the absorber. The actual hardware requirements followed from the philosophy that the absorber should:

a) be portable and capable of installation at the centre of the lower deck,

b) exhibit no modes of vibration other than the desired vertical bounce made in the neighbourhood of 12 Hz,

c) be adjustable in frequency between 11 Hz and 14 Hz.

The concept for the absorber changed several times before being finalized. Use was made of the PAL2 finite element analysis package to evaluate the static stresses and natural frequencies and mode shapes. The final configuration shown in Figure 17 featured a pin jointed beam supported on a rigid frame which attached directly to the stanchion floor brackets in Car 2030, the test car. This enabled the effect of stanchions and absorber to be assessed simultaneously or individually. The mass of the absorber was made up of eleven weights of 66.8 kg average each which could be easily added or removed to adjust the absorber frequency. Because the central absorber beam deflected parabolically the mass located near the beam ends was less effective at providing a counter force then that at the centre and probably reduced the effectiveness of the absorber by 20-30%. However, this design isolated other modes of vibration of the masses to such an extent that only the primary vertical bounce mode was present in the frequency range up to 35 Hz. For example, with 9 weights in place the first four modes from PAL2 were:

TABLE 4.3-2
Absorber Predicted Modes

Mode #	Frequency - Hz	Mode Shape
1	13.31	1st vertical bounce of absorber beam (2 nodes)
2	40.25	1st lateral bending of absorber beam (2 nodes)
3	46.49	2nd lateral bending of absorber beam (2 nodes)
4	63.11	2nd vertical bounce of absorber beam (3 nodes)

The variation of frequency of mode 1 with absorber mass is shown in Figure 18 compared to a bench test performed by MTO on 1988 May 19. The decay response to an impact hammer blow was recorded and analysed for frequency at three mass settings of the absorber (5, 7 and 9 weights applied). Data was gathered in the form of decay plots Figure 19, and transfer function plots, Figure 20, between impact force and response velocity at the beam centre. The results showed that the absorber frequency lay below the predicted values by less than 10% indicating that the actual beam wall thickness or geometry was slightly undersized compared to the values assumed in the PAL2 program.

4.4 SECONDARY DAMPER BUSH MOD ON BASELINE CAR

The decision to pursue modification of the secondary vertical damper mounting bushings (rubber) was taken after reviewing CCRI test work in 1980 (3) and (13). At that time, Hawker Siddeley Canada Ltd. (HSC) conducted the tests to investigate the possibility that dither was excited by truck and bolster vertical motion. The solution, if this was true, was to reduce the transmissibility of the secondary vertical shock absorbers by reducing the effective bushing stiffness or reducing the damper rate. The results of these tests generally showed that dither could be reduced by substantial amounts (70%) with a bevelled bushing arrangement (Configuration 3-2 of Test on 1980 August 07 reported in (3)). Concern regarding the adverse effect of the bushing mod upon carbody roll behaviour was not

supported by subsequent testing (4), however, the occasional banging noise from unloading of the washers on either side of the bushings and the effect of such a mod upon bushing fatigue life dampened the initial enthusiasm for a production version of the mod. From our perspective, it was important to include this approach to solving the dither problem as it did potentially represent a simple cost effective, light weight solution. Thus it was short listed for theoretical evaluation along side the other modifications described in this report.

To evaluate the effects of the modification upon Bi-level ride quality a series of simulation runs was made with a range of bushing stiffnesses. Before describing the runs, it is worthwhile to look at the forces and displacements range encountered under normal operating conditions.

For normal track conditions where dither is not excited, the carbody acceleration is typically within ±0.021 g at 1.5 Hz thus:

$$V = \text{peak damper velocity} = \frac{a}{(2\pi f)} = \frac{0.021\ g}{(2\pi 1.5)} = 2.2 \times 10^{-2}\ \text{m/sec}$$

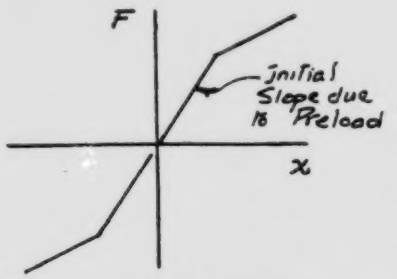
for a 152.4 mm arm on the Houdaille RC damper this gives a peak angular velocity of:

$$\Omega_{max} = \frac{V}{l} = \frac{0.022}{0.1524} = 0.14\ \text{rad/sec.}$$

From the Houdaille RC Catalogue, peak torque for the factory setting is 800 in. lb (90.4 Nm) and peak link force is 133 lb (592 N). Similarly, for a car excited in dither, the truck motion is assumed to dominate and from (13) if the truck acceleration is ±2 g at 13 Hz, then the truck displacement is ±2.9 mm and the Houdaille "RC" link force required is ±2400 lb (10,680 N). This is considerably more than required by the no dither case above and gives the range in which the resilience of the bushing needs to be reduced (±7.9 mm). Another design constraint emerging from the need to keep the noise down and ensure adequate fatigue life is to have a preload sufficient to contain normal background vibration. This would suggest a minimum 133 lb (592 N) preload although (13) recommended 750 lb (3338 N) to allow for higher initial disturbance levels. The requirement for a preload conflicts with the need to have a low initial resilience in the bushing assembly. A decision on the level

of preload was made after the results of the simulation, however, the above preload and displacement limits suggested a range of bushing resilience to use in the simulation. This range was from

$$\frac{592}{0.0029} = 2.04E5 \text{ N/m (1166 lb/in) to } \frac{3338}{0.0029} = 1.15E6 \text{ N/m (6577 lb/in)}$$



As far as the simulation is concerned, it represents only linear elements and thus, a) models no particular damper bushing arrangement and b) models only the initial stiffness of the bushings. If preload is present then initial stiffness of the assembly is higher for a given bushing stiffness and is dependent upon the r.m.s. amplitude applied to the assembly. Thus Model BILEVEL6 was run with a series spring and damper at each shock location for five values of bushing assembly stiffness, K_c, ranging from 36.8 MN/m (unmodified bushing) down to 0.01 MN/m. The effects upon upper and lower deck accelerations in the 12.5 Hz third octave band were studied. The results in Figure 21 indicate that a rapid decrease in dither vibration occurs for bushing stiffness, K_c less than 1 MN/m which levels off at 0.1 MN/m. Thus 0.1 MN/m (570 lb/in.) appears to be the optimum value for K_c. The runs made out with BILEVEL6 are summarized below:

TABLE 4.4-1
Predicted Car Response to Secondary Bushing Stiffness

Case #	Secondary Vertical Damping Ratio	Load	V m/s	Track Class	Secondary Link K_c - MN/m	r.m.s. Response at 12.5 Hz					
						Acceleration		Displacement		Velocity	
						Upper Deck (18) g	Lower Deck (19) g	Link f/LHS (45) m	Link f/RHS (46) m	Damper f/LHS (49) m/s	Damper f/RHS (50) m/s
1	35.9%	AWO	35.76	6	36.81	0.1467	.064	2.59E-5	2.63E-5	1.70E-2	1.73E-2
2	35.9%	AWO	35.76	6	1.22_7	0.1541	.0684	2.86E-4	8.03E-4	1.72E-2	1.76E-2
3	35.9%	AWO	35.76	6	0.122_7	0.0660	.0291	1.40E-3	1.75E-3	3.07E-3	3.84E-3
4	35.9%	AWO	35.76	6	0.010	0.0589	.0264	1.39_9E-3	1.720E-3	2.43E-4	3.02E-4
5	35.9%	AWO	35.76	6	0.400	0.1009	.0455	1.22E-3	1.37E-3	8.72E-3	9.80E-3

( ) Bracketed numbers refer to body number in model.

From this data, a required force-deflection characteristic was derived in Figure 22. The initial stiffness of 0.1 MN/m ends at a displacement of 2.9 mm, then the stiffness increases to 16 times the initial value (1.6 MN/m). This means that the damper will transmit only 290 N (65 lb) compared with 10,680 N (2400 lb) with the unmodified arrangement.

To achieve this characteristic would have required a joint design, analysis and test program for which there was insufficient time although the test fixture EX215-600-002 was built to develop the Houdaille bush modification. In lieu of this, the original HSC configuration 3-2 from (13) was adopted for the test program on car 2030 (Koni shock equipped) and is described in Volume II.

4.5 IMPROVED DEFINITION OF PROBLEM REFINED BASELINE

After poor results from the first car modification, stanchions, it was decided to look further into the behaviour of the truck and its effect on dither vibration. To support this, a special test entitled the "Improved Definition of Problem" test was carried out in 1988 June on car 2030. From the data collected (see Appendix D) at twelve locations (20 channels), a new picture of car behaviour was obtained which was used to refine the dynamic model, BILEVEL3. What was evident from this testing was that:

a. Wheel rpm did not correlate strongly with the key frequencies in the truck.

b. The rails show increased energy in the neighbourhood of 12-15 Hz.

c. The wheelsets show peaks in vertical motion of 5 Hz and 17 Hz.

d. The ends of the truck frame heave strongly at 17 Hz with some motion at 5 and 12 Hz.

e. The bolster heaves strongly at 12.5 Hz.

f. Longitudinal motion of bolster occurs at 12.5 Hz and is low in comparison to vertical direction.

g. Truck yaw motion occurs at 4.5 and 17 Hz.

Of the above findings, the most significant was the distinct difference between truck frame and bolster frequency peaks which suggested that mechanisms other than rigid body motion on suspension components were involved. This led to separate static deflection tests being conducted by Dofasco on the bolster and truck frame (See Appendix E).

The process of refining the dynamic model, was to start with BILEVEL3 and use the above tests results to tune the suspension components while trying to match the measured response on the body of car 2030 used in earlier model development. All runs were made at 80 mph for the car at AW0. The original Dofasco suspension parameter values [(8) + Appendix A] were kept in mind in order not to drift beyond reasonable limits during adjustment of the model. The first series of model adjustments were done in the BILEVEL7 series of runs where the BILEVEL3 three-mass body model in Figure 3 was exercised. After eight runs with this model, the results did not show the expected characteristics even after a wide variation in parameters such as the secondary damper link bush stiffness, sidebearer stiffness and chevron stiffness. It was difficult to obtain a clean 12 Hz bounce on the bolster without some 17 Hz "leaking" through from the frame. Because this model did not differentiate between intermediate deck and lower deck centre, it was decided to reuse the three mass representation of the body so that two half body masses with c.g.s. located near the door centres were joined with a six degree of freedom spring at the car centre as shown in Figure 23. With a small mass representing the upper deck moving mass connected to the half-body masses, the BILEVEL8 model series was started. The purpose of this model was to represent the first bending made of the body which occurs at 12 Hz and thus improve the prediction of the intermediate deck response. The model runs are shown in Table 4.5-1.

The first task was to adjust the interbody pitch spring stiffness and damping to match the correct response magnitude on the upper, lower and intermediate decks. It was found that 6% critical damping for both the inter-car pitch damper and upper deck vertical heave damper gave the best results. At this point a 5% discrepancy in the lower deck centre acceleration was noticed when it was calculated separately from the motion on each half body. One would expect identical results as there is fore-aft symmetry of the car and no uni-directional effects. It was a result of an out of phase pitch

TABLE 4.5-1 IMPROVED DEFINITION SIMULATION RUNS

DATE	TIME	CASE	PSM	$K_{CHEVRON}$ N/m	$K_{SIDE-BEARER}$ N/m	SEC'DY SUSP'N FREQ Hz	SEC'DY SUSP'N ζ %	BODY BENDING $f_{BENDING}$ Hz	BODY BENDING $\zeta_{BENDING}$ %	UPPER DECK f_{UD} Hz	UPPER DECK ζUD %	g-rms CAL'D RESPONSE AT 12.5 Hz U.Deck(18)	L.DECK(19)	I.DECK(5)	E/V's	REMARKS
1988 AUG 25	16:15	801	RIDE RPSD	7.09E6	3.08E6	1.355	35.9	12.1	0.5	13.4	25	1.28E-01 0.098E-01	0.524E-01 1.00E-01	0.27E-01 0.817E-01		Body Bending Mode Freq. hi
AUG 26	9:04	802	RIDE	—	—	—	—	10.875	0.1	—	—	NA	NA IF 16	NA IF 17		Body Bending at111 hi
AUG 26	14:48	803	RIDE RPSD	—	—	—	—	9.1	—	—	—	1.65E-01	1.58E-01	1.01E-01		Body Bending 7, 12 Hz
OCT 04	17:16	804	RIDE RPSD	—	—	—	—	—	0.7	—	—	1.30E-01	1.26E-01	0.80E-01		0.7% Intercar ζ
OCT 06	16:09	805	—	—	—	—	—	—	5.0	—	—	0.569E-01	0.625E-01	0.341E-01		5.0% Intercar ζ
OCT 07	9:55	806	—	—	—	—	—	—	7.5	13.4	10	0.739E-01	0.507E-01	0.227E-01		7.5% Intercar ζ ; 10% UD ζ
OCT 11	8:57	807	—	—	—	—	—	—	—	—	6	0.923E-01	0.511E-01	0.215E-01		6% Intercar; 2 Deck strange IF16
OCT 12	9:04	807a	—	—	—	—	—	—	—	—	—	—	—	—		Printed IF's from (5) + (19) to
OCT 12	11:35	807b	—	—	—	—	—	—	—	—	—	—	—	—	Body 18 long spring at 0, -2.057	compare body c.g. accn
OCT 12	12:55	808	—	—	—	—	—	—	—	—	—	—	—	—	Pitch Mode at111 not proper.	Tried sharing spring /dampers of (18) with (5) & (19)

TABLE 4.5-1 (Cont'd)

DATE	TIME	CASE	PGM	$K_{CHEVRON}$	$K_{SIDE-BEARER}$	SEC'DY SUSP'N		BODY BENDING		UPPER DECK		g-rms CAL'D RESPONSE AT 12.5 Hz				REMARKS
						FREQ		$f_{BENDING}$	$C_{BENDING}$	f_{UD}	UD					
				N/m	N/m	Hz	%	Hz	%	Hz	%	U.Deck(18)	L.DECK(19)	I.DECK(5)	E/V's	
1988																
OCT 13	8:11	809	RIDE RPSD	7.09E6	3.08E6	1.355	35.9	9.1	7.5	13.4	6	0.923E-01	0.511E-01	0.215E-01		Tried reducing interbody shear spring k by 16 to reduce round-off error.
OCT 13	16:05	810	|	|	|	|	|	|	|	|	|	|	|	|	|	Tried interchanging body nos 5 - 19 to see if 1.57 Hz mode fixed.
OCT 17	8:56	811	|	|	|	|	|	|	|	|	|	|	|	|	|	Added long spring to axle 1 . Printed off body acc'ns.
OCT 21	9:20	812	|	|	|	|	|	|	|	|	|	0.921E-01	0.49E-01	0.214E-1		Added long spring to axle #4.
OCT 21	13:39	813	|	|	|	|	|	|	|	|	|	|	|	|	|	Stiffer Interbody Vert Shear, k
NOV 02	10:07	814	|	|	|	|	|	|	|	|	|	|	|	|	|	Stiffer Interbody Long Shear, k

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TABLE 4.5-1 (Cont'd)

DATE	TIME	CASE	PDH	$K_{CHEVRON}$ N/m	$K_{SIDE-BEARER}$ N/m	SEC'DY SUSP'N FREQ Hz	SEC'DY SUSP'N %	BODY BENDING $f_{BENDING}$ Hz	BODY BENDING $\zeta_{BENDING}$ %	UPPER DECK f_{UD} Hz	UPPER DECK ζ_{UD} %	g-rms CAL'D RESPONSE AT 12.5 Hz U.Deck(18)	g-rms L.DECK(19)	g-rms L.DECK(5)	E v a	REMARKS
1988		Ref														
NOV 02	14:48	815	RIDE RPSD	7.09E6	3.08E6	1.28	21.9	11.3	10.4	15.31	10.8	1.28E-1 0.23E-1	0.52AE-1 0.3E-1	0.27E-1 0.12E-1		1/2 body c.g.'s over airbag loc'n.
NOV 04	8:26	816		—	—	—	—	—	—	—	—	—	—	—		Lat. body-bolster rod flattened
DEC 15	16:50	817		—	—	—	—	12.4	10.2	14.5	10.6	1.02E-1	0.50E-1	0.48E-1	0.22E-1	Restore cg to ±5.79% No longit. Dof. Virtually identical to 814.
DEC 16	10:32	818		—	14E6	1.35	27.7	12.4	9.71	14.5	10.7	1.47E-1	0.71E-1	0.69E-1	0.33E-1	DOFASCO Frame/ Bolster Effects incl'd. Car As-is.
DEC 16	17:02	819		—	—	1.15	30.9	—	9.77	—	—	1.53E-1	0.74E-1	0.72E-1	0.34E-1	V. Stiff Truck & Bolster Frame
1989 JAN 06	17:31	820		3.54E6	—	1.33	25.6	12.3	11.9	—	11.4	0.81E-1	0.41E-1	0.39E-1	0.19E-1	Truck frame bounce reset to 16.6 Hz & Damping in frame and bolster bending set to 2%.
JAN 10	16:06	821		1.18E7	—	1.36	28.6	12.5	9.6	14.6	10.5	1.56E-1	0.75E-1	0.73E-1	0.35E-1	Return to Case 818 & Stiffer Chevron
JAN 11	10:34	822		1.52E7	—	1.34	29.0	12.6	9.7	—	10.7	1.47E-1	0.73E-1	0.70E-1	0.33E-1	Stiffer chevrons to give 16.0 Hz frame bounce.

TABLE 4.5-1 (Cont'd)

DATE	TIME	CASE	PGM	$K_{CHEVRON}$ N/m	$K_{SIDE-BEARER}$ N/m	SEC'DY SUSP'N FREQ Hz	SEC'DY SUSP'N %	BODY BENDING $f_{BENDING}$ Hz	BODY BENDING $C_{BENDING}$ %	UPPER DECK f_{UD} Hz	UPPER DECK UD %	g-rms CAL'D RESPONSE AT 12.5 Hz U.Deck(18)	g-rms CAL'D RESPONSE AT 12.5 Hz L.DECK(19)	g-rms CAL'D RESPONSE AT 12.5 Hz L.DECK(5)	g-rms CAL'D RESPONSE AT 12.5 Hz I.DECK(5)	E v e	REMARKS
1989																	
JAN 11	13:06	823	RIDE RPSD	1.52E7	14E6	1.34	29.0	12.8	11.6	14.4	13.3	1.28E-1	0.66E-1	0.62E-1	0.27E-1		Intercar now 7.5%
JAN 12	10:18	824	|	|	|	|	|	|	|	|	|	1.33E-1	0.66E-1	0.63E-1	0.32E-1		1/2 Body c.g. shifted to 5.478 m
JAN 12	16:05	825	|	|	8.5E6	1.35	27.4	|	|	14.2	12.9	1.33E-1	0.67E-1	0.63E-1	0.32E-1		Case 317 sidebearers
JAN 13	13:49	826	|	|	5E8	1.37	30.8	13.0	11.6	14.4	14.8	1.49E-1	0.76E-1	0.71E-1	0.35E-1		V. stiff sidebearers to see if frame bending alone gives 12 Hz.
JAN 13	18:05	827	|	5.89E6	2.41E6	1.26	19.6	12.6	13.4	14.1	11.8	0.97E-1	0.60E-1	0.60E-1	0.35E-1		Chevrons & side bearers softened - gave better result
JAN 16	16:38	828	|	7.29E6	|	1.25	19.9	12.7	12.9	14.0	11.1	0.93E-1	0.42E-1	0.40E-1	0.20E-1		Chevrons stiffened to improve 16.6 Hz mode
JAN 17	11:05	829	|	|	|	1.24	20.0	12.4	8.4	14.5	8.4	1.20E-1	0.50E-1	0.49E-1	0.27E-1		6% Interior Damping
JAN 17	16:14	830	|	|	|	|	|	|	8.5	14.4	8.2	1.56E-1	0.65E-1	0.64E-1	0.34E-1		Rail stiffness reduced to 1.0E6. Gives bounce at 12.5 Hz.
JAN 19	10:31	831	|	|	3E8	2.16	|	12.5	|	14.5	|	2.40E-1	0.99E-1	0.99E-1	0.56E-1		Extreme Case - stiff sidebearers.

TABLE 6.5-1 (Cont'd)

DATE	TIME	CASE	PSM	$K_{CHEVRON}$	$K_{SIDE-BEARER}$	SEC'DY SUSP'N		BODY BENDING		UPPER DECK		g-rms CAL'D RESPONSE AT 12.5 Hz				E v a	REMARKS
						FREQ		$f_{BENDING}$	$C_{BENDING}$	f_{UD}	UD	U.Deck(18)	L.DECK(19)	L.DECK(5)	I.DECK(5)		
				N/m	N/m	Hz	%	Hz	%	Hz	%						
1989 FEB 01	9:50	832	RIDE RPSD	7.29	2.0	1.23	18.6	12.4	8.4	14.5	8.4	1.08E-1	0.45E-1	0.45E-1	0.24E-1		Softer K_{SB} than #839
FEB 02	13:19	833	—	5.56	14.0	0	36.8	12.4	10.2	14.3	9.8	1.44E-1	0.63E-1	0.61E-1	0.34E-1		Baseline for improvement
FEB 03	13:26	834	—	—	—	—	1.0	12.5	8.7	14.5	10.4	2.10E-1	0.94E-1	1.09E-1	0.54E-1		K_{BUSH} = 1.0 MN/m
FEB 03	14:10	835	—	—	—	—	0.1	12.7	8.7	14.5	10.2	1.07E-1	0.65E-1	0.46E-1	0.33E-1		K_{BUSH} = 0.1 MN/m
FEB 03	15:11	836	—	—	—	—	0.01	12.7	8.8	14.5	10.2	0.959E-1	0.58E-1	0.39E-1	0.30E-1		K_{BUSH} = 0.01 MN/m
FEB 03	16:17	837	—	—	—	48,690	36.8	12.6	10.3	14.4	10.2	1.04E-1	0.66E-1	0.44E-1	0.24E-1		C_{PRI} = 48,690
FEB 06	8:02	838	—	—	—	32,460	—	12.6	10.3	14.4	10.1	1.09E-1	0.68E-1	0.44E-1	0.25E-1		C_{PRI} = 32,460
FEB 06	9:17	839	—	—	—	48,690	0.01	12.7	9.2	14.5	10.3	0.41E-1	0.18E-1	0.17E-1	0.095E-1		K_{BUSH} = 0.01 MN/m C_{PRI} = 48,690
FEB 06	10:08	840	—	—	—	0	36.8	36.0	N.A.	13.5	4.7	0.83E-1	0.37E-1	0.36E-1	0.14E-1		K interear pitch x 10
FEB 06	11:03	841	—	—	—	0	36.8	12.4	10.1	14.3	9.8	1.48E-1	0.64E-1	0.63E-1	0.35E-1		K_{BOLST}, K_{FRAME} x 10
JAN 15	10:30	842	—	—	—	0	—	30	N.A.	30	N.A.	0.137E-1	0.38E-1	0.38E-1	0.15E-1		K_{BOLST}, FRAME, BODY, U/DECK x 10

mode in which considerable relative vertical displacement occurred at the joint between the half bodies #5 and #19 which did not appear to be physically achievable in view of the stiff connection there. This mode was investigated with a simpler model involving the three mass carbody only on the RIDE software as well as on a commercial FEA package, PAL2. The results supported the existence of this anti-symmetric pitch mode thus proving it to be a generic property of this representation.

During investigation of the antisymmetric pitch mode, the BILEVEL9 model series was created to look at the effect of locating the centre of gravity of each half body at the car centre. Thus each half-body resembled a uniformly distributed mass running the length of the car and pinned to the other mass at the car centre. This configuration shown in Figure 23 was termed the "scissors" arrangement. While it did not exhibit the anti-symmetric pitch mode, it exaggerated the intermediate deck response and hence could not be used in further work.

The second task was to install bolster and frame vertical bending effects. This was done similarly to the BILEVEL3 model where the bolster was divided into two equal masses one on top of the other. The top was called the bolster bending mass was given ty, tz and Rx (roll) freedoms while the bottom mass called the "bolster" was given ty, tz, Rx (roll), and Ry (pitch) degrees of freedom. A similar approach was used for the truck frame. The vertical springs connecting the bending mass to the frame and bolster masses were sized from the Dofasco test (Appendix 5) and the damping selected to lie at 1.4% critical. Runs were made (818 & 819) with the bending effects in place and then removed. The effect upon ride response was to produce a less than 5% decrease when bending effects were included. However, while the bolster showed its proper 12 Hz bending behaviour, the truck frame did not and appeared to move at the same frequencies as the bolster. Three runs (820 to 822 incl.) were made to investigate the sensitivity of frame heave frequency to chevron primary spring stiffness. This stiffness ranged from 3.5E6 N/m to 1.5E7 N/m where the highest value produced a frame bounce of 16 Hz but because of the amount of coupling from the sidebearers (using the Dofasco recommended value of 14E6 N/m per side) the bolster frequency also increased. To remedy this, the sidebearer stiffness was reduced to the value used in Run 317, the initial baseline model. This (run 825) helped to introduce some

decoupling of the bolster at 12 Hz but not enough to match the desired behaviour. At this point, it was logical to test the idea that bolster and frame bending effects were responsible for the distinctive 12 Hz bounce behaviour of the bolster. Run 826, with stiff chevrons and very stiff sidebearers attempted to use the bending stiffness of the bolster and frame to generate a 12 Hz bounce mode on the bolster alone while the frame bounced at 16 Hz. The results showed that three modes at 15.2, 15.7, 13.0 Hz were produced which had nearly equal participation of bolster and frame; meaning that the bolster was coupled more strongly to the frame than the carbody. It was felt at this stage, that to properly separate the bolster mode from frame vertical modes, we had to size the chevrons to cause a 16 Hz bounce on the frame mass above and to reduce the sidebearer stiffness per side to decouple the bolster. Run 827, with this idea, gave the most promising results so far with two modes at 15.1 and 15.2 Hz on which frame motion was twice that of the bolster while the bolster exhibited a mode at 12.6 Hz in which the bolster motion was twice that of the frame. The chevrons were stiffened by (16.6/15.3)2 in Run 828 to further improve frame bounce frequency. While the model frequencies were closer to desired values, the ride response at 12.5 Hz was low and thus needed to be boosted by 30%. This was accompanied by decreasing the inter-body damping to 5.2% critical (it had been previously increased to 10%) in Run 829. The effect of softening rail support stiffness from 2.8E7 N/m to 1.8E7 was studied in Run 820. This had the effect of raising all response levels and lowering frame and bolster bounce frequencies by $\sim$1 Hz. The initial value of 2.8E7 N/m was chosen because it represents nominal ballast track conditions and has been used unchanged for all response calculations. Run 831 which is essentially like run 819 was a final check on the significance of frame bending. As expected, it did not give proper modal behaviour for the bolster and truck frame.

The best match to our required behaviour is run 829, however, in tuning the model we have used values for the chevrons which are 3.7 times stiffer than the original Dofasco values and sidebearers which are 5.8 times softer than the Dofasco values. For the modification studies in sections 4.6 through 4.8, it was decided to revert to the Dofasco sidebearer stiffness values (8) and assign a (lower) chevron stiffness sufficient to give a 12.0 Hz frame plus bolster bounce frequency. The sidebearer stiffness is likely to

increase with time over the initial value as is the chevron stiffness. Of the two modes, the 12 Hz bolster bounce is most likely to influence upper deck dither. The 17 Hz frame mode could, in fact, be non-existent at the sidebearer pads and hence not be transmitted up to the body. This could not be verified in the 1988 June tests as no vertical accelerometer instrumentation was placed at the centre of the truck frame. Thus Run 833 was used as the refined baseline for subsequent runs. A listing of the parameters used in Run 833 is given in Appendix F.

4.6 PRIMARY DAMPER MOD ON REFINED BASELINE CAR

The refined baseline model from run 833 in section 4.5 was configured to model a primary damper attached between each axle box and the truck frame as done previously in (10).

This simulation was rerun because it was felt that with the changes to the truck suspension parameters in the refined model, the effect of the primary dampers may be different than found in (10). To see if this was true, two cases were run as follows:

TABLE 4.6-1

Run #	Primary Damper Rate - Ns/m	% Critical Damping on Truck Frame Bounce	% Houdaille Factory Setting
837	48.69E3	45%	125%
838	32.46E3	30%	85%

The results are shown in Figure 25 which compares the upper deck vertical acceleration at 12.5 Hz for each case above with the baseline value. As in the previous study (10), the dither response falls off most rapidly for damping between 0% and 30% then levels out at a point between 45% and 60% critical damping (based upon truck frame heave). Thus there is little improvement between case 837 and 838. Compared to the results in (10), the improvement is larger here. At 30% critical damping, a 24.1% reduction is realized while at 45%, the improvement is 27.6%. The means that the primary dampers could make a noticeable ride improvement.

4.7 SECONDARY DAMPER BUSH ON REFINED BASELINE CAR

The secondary vertical damper link bush mod described in Section 4.4 was evaluated on the refined baseline model to see if any changes to its effectiveness occurred. Four cases were run as described below including one in which the bush mod was combined with the primary damper mod:

TABLE 4.7-1

Case	Damper Bush Stiffness MN/m	Primary Damper Rate Ns/m	Upper Deck Acc'n.- g.r.m.s.
834	1.0	0	0.218
835	0.1	0	0.107
836	0.01	0	0.0096
839	0.01	48,690	0.0041

The results are shown in Figure 25 in the form of a bar chart and Figure 26 compared to the initial baseline model predictions. A smaller improvement is noticeable relative to the unmodified damper bush with the refined model even though the response level is the same between the two models.

The combined case 839 gives the second largest improvement and suggests that the primary dampers in conjunction with the damper bush mod could give the most cost effective solution if the bush stiffness could be kept to a small value (0.01 MN/m) over the initial ±3 mm of working displacement.

4.8 STRUCTURAL STIFFENING MODS

Having obtained the refined baseline model in which the 1st bending mode of the body as well as the frame and bolster bending were represented, it seemed appropriate to look at the gross effects upon ride of a general stiffening of the car structure.

To do this runs were made in which first the carbody bending stiffness was increased by a factor of ten, then the Dofasco measured truck and bolster bending stiffnesses were increased by a factor of ten as shown below:

Case	Carbody Bonding Stiffness Nm/rad	Truck Frame Stiffness N/m	Bolster Stiffness N/m	Upper Deck Stiffness N/m
840	1.75E10	9.83E7	1.58E8	3.8E6
841	1.75E9	9.83E08	1.58E9	3.8E6
842	1.75E10	9.83E08	1.58E9	3.8E7

Note: Underlined values are baseline.

From runs 840 and 841 body bending stiffness has the largest effect at reducing the upper deck dither as soon in Figure 25. This is not surprising as the frame/bolster bending effect appeared to be small during refined baseline runs 818 and 819.

The bolster and frame bending stiffnesses are much higher than the surrounding suspension component stiffnesses and thus produce smaller displacements. A combined run 842 was made where the truck frame, bolster, body and upper deck were stiffened by ten times simultaneously. The results indicate that a 10 times reduction in upper deck response is possible; a larger improvement than produced by any of the other modifications.

5.0 FINITE ELEMENT MODEL RESULTS

In parallel with the work described in Section 4.0, the Engineering group at CCW, Thunder Bay undertook to develop a finite element model of the car to use in evaluating certain proposed mods (7). These mods required a detailed representation of the car structure as a basic working tool in order to look at the natural modes and frequencies. If the offending modes in the vicinity of 12-15 Hz could be shifted to higher frequencies or made benign then the overall vertical ride quality of the car would improve. The end result of this analysis was a relative ranking of the effectiveness of each modification using a pseudo acceleration concept described later.

The baseline car model was an updated version of one made at CCW in 1983. Changes to lower, intermediate and upper deck floor panel modulus of elasticity; element mass density (to achieve current AWO car shell weight); revised centre sill section properties and raising of lower floor waterline were incorporated.

The accuracy of the model was checked in an impact test on a partially finished carshell at CCW in 1987 October 01. Good correlation was found with predicted frequencies of the first five modes thus justifying its use in the modification studies.

5.1 BASELINE CARBODY

The baseline model (10) was made with an AWO weight of 9365 kg (20,650 lb) and run to extract the first five modes of vibration. Of these, only modes two and three had any significance to the dither problem. The modes were:

Mode #	Frequency - Hz	Shape
2	12.1	1st Body Bending
3	15.5	Upper Deck Heave

These are illustrated in Figures 27 and 28 from (10). The results are also tabulated in Tables 5.1-1 through 5.1-8 where model frequency, relative displacement, pseudo acceleration [= relative displacement x (2 x frequency)^2] and effectiveness indicator. The latter is equal to (sum from node 1 to node 4 pseudo accelerations)$^{-1}$; a higher value indicates a more effective

modification. Nodes 1 to 4 are located on the upper deck centreline from the car mid-point outwards. Comparison of effectiveness indicators provides a relative rating of proposed modifications. Tables 5.1-1 to 5.1-8 are organized into two groups, pseudo acceleration with respect to sidewall at the same waterline and station as the node points of interest and pseudo acceleration with respect to the secondary suspension connection node (266). Within each group is a pair of tables for upper floor and a pair for lower floor. Within each pair are the results for mode 2 (body bending) and mode 3 (upper deck heave). Both modes collectively comprise the dither vibration.

5.2 INCORPORATION OF STIFFENED FLOOR PANELS

When the Bi-level Ride Improvement program was first discussed, one of the first proposals to eliminate dither was to stiffen the upper deck floor panels using a composite aluminum balsa sandwich construction. The equivalent thickness and modulus of elasticity of the panel were calculated and used in a plate element model for the upper deck floor between stations 347.5 and 510.0. The results of running this mod are tabulated in Tables 5.1-1 through 5.1-8 under "Upper Floor Panel". The corresponding mode shapes are given in Figures 29 and 30.

5.3 INCORPORATION OF STANCHIONS

Stanchions were first proposed as a modification after a short test at CCW in 1987 July. At that time, it was thought that the major part of dither came from flexing of the upper deck central area. It was discovered that a substantial reduction in the motion of the upper deck at 14 - 17 Hz occurred when struts were placed between the upper and lower decks. Because of uncertainties on the minimum number of stanchions required to quiet the dither vibration three arrangements of four, six and eight stanchions were proposed. Only two of these could be tried, the four and eight stanchion arrays, because the FEA model was for a $\frac{1}{4}$ car only. The stanchions were designed as 1" schedule 40 stainless pipe and required four new nodes to be added to the model.

TABLE 5.1-1 - UPPER FLOOR 2ND MODE PSEUDOACCELERATION WITH RESPECT TO SIDEWALL

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY $\omega = 2\pi f$	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION $A = \omega^2 D/386.4$ AT NODE				EFFECTIVENESS INDICATOR *
			15	31	63	95	15	31	63	95	
AWO, No Modification (Control)	12.052	75.72	0.102	0.087	0.073	0.043	1.51 g	1.29 g	1.08 g	0.64 g	0.22
Upper Floor Panel	12.187	76.57	0.051	0.047	0.043	0.036	0.77 g	0.71 g	0.65 g	0.55 g	0.37
Four Stanchions	12.041	75.66	0.072	0.061	0.067	0.043	1.07 g	0.90 g	0.99 g	0.64 g	0.28
Eight Stanchions	12.033	75.61	0.074	0.062	0.052	0.041	1.09 g	0.92 g	0.77 g	0.61 g	0.29
1/8" Shear Plate Above Door Opening	12.217	76.76	0.104	0.089	0.074	0.044	1.59 g	1.36 g	1.13 g	0.67 g	0.21
Boxed in Upper Floor Beams	12.078	75.89	0.059	0.047	0.041	0.032	0.88 g	0.70 g	0.61 g	0.48 g	0.37

* Effectiveness Indicator = $\left| \frac{g}{\sum \text{pseudoacceleration}} \right|$

The value closest to unity indicates the most effective modification.

TABLE 5.1-2 - UPPER FLOOR 3RD MODE PSEUDOACCELERATION WITH RESPECT TO SIDEWALL

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY $\omega = 2\pi f$	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION $A = \omega^2 D/386.4$ AT NODE				EFFECTIVENESS INDICATOR *
			15	31	63	95	15	31	63	95	
AW0, No Modification (Control)	15.530	97.57	-0.435	-0.395	-0.288	-0.082	-10.72 g	-9.73 g	-7.10 g	-2.02 g	0.03
Upper Floor Panel	15.696	98.62	-0.313	-0.301	-0.226	-0.077	-7.88 g	-7.58 g	-5.69 g	-1.94 g	0.04
Four Stanchions	15.871	99.72	-0.040	-0.052	-0.192	-0.083	-1.03 g	-1.34 g	-4.94 g	-2.14 g	0.10
Eight Stanchions	15.963	100.30	-0.069	-0.060	-0.049	-0.065	-1.80 g	-1.56 g	-1.28 g	-1.7 g	0.16
1/8" Shear Plate Above Door Opening	15.530	97.57	-0.436	-0.395	-0.289	-0.082	-10.72 g	-9.73 g	-7.10 g	-2.02 g	0.03
Boxed in Upper Floor Beams	15.754	98.98	-0.274	-0.247	-0.192	-0.064	-6.95 g	-6.26 g	-4.87 g	-1.62 g	0.05

* Effectiveness Indicator = $\left| \frac{g}{\sum \text{pseudoacceleration}} \right|$

The value closest to unity indicates the most effective modification.

TABLE 5.1-3 - LOWER FLOOR 2ND MODE PSEUDOACCELERATION WITH RESPECT TO SIDEWALL

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY = 2 f	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION A = ^{2}D/386.4 AT NODE			
			1	17	49	81	1	17	49	81
AWO, No Modification (Control)	12.052	75.72	0.051	0.049	0.041	0.023	0.76 g	0.73 g	0.61 g	0.34 g
Upper Floor Panel	12.187	76.57	0.054	0.052	0.043	0.024	0.82 g	0.79 g	0.65 g	0.36 g
Four Stanchions	12.041	75.66	0.055	0.052	0.042	0.024	0.81 g	0.77 g	0.62 g	0.36 g
Eight Stanchions	12.033	75.61	0.055	0.053	0.044	0.024	0.81 g	0.78 g	0.65 g	0.36 g
1/8" Shear Plate Above Door Opening	12.217	76.76	0.052	0.049	0.041	0.025	0.78 g	0.75 g	0.63 g	0.38 g
Boxed in Upper Floor Beams	12.078	75.89	0.053	0.049	0.042	0.023	0.79 g	0.73 g	0.63 g	0.34 g

TABLE 5.1-4 - LOWER FLOOR 3RD MODE PSEUDOACCELERATION WITH RESPECT TO SIDEWALL

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY $\omega = 2\pi f$	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION $A = \omega^2 D/386.4$ AT NODE			
			1	17	49	81	1	17	49	81
AWO, No Modification (Control)	15.530	97.57	-0.001	-0.001	-0.005	-0.008	-0.02 g	-0.02 g	-0.12 g	-0.20 g
Upper Floor Panel	15.696	98.62	-0.011	-0.010	-0.012	-0.012	-0.28 g	-0.25 g	-0.30 g	-0.30 g
Four Stanchions	15.871	99.72	-0.048	-0.044	-0.032	-0.018	-1.24 g	-1.13 g	-0.82 g	-0.46 g
Eight Stanchions	15.963	100.30	-0.054	-0.052	-0.041	-0.024	-1.451g	-1.35 g	-1.07 g	-0.62 g
1/8" Shear Plate Above Door Opening	15.530	97.57	-0.001	-0.001	-0.005	-0.008	-0.02 g	-0.02 g	-0.12 g	-0.20 g
Boxed in Upper Floor Beams	15.754	98.98	-0.012	-0.011	-0.013	-0.012	-0.30 g	-0.28 g	-0.33 g	-0.30 g

TABLE 5.1-5 - UPPER FLOOR 2ND MODE PSEUDOACCELERATION WITH RESPECT TO SUSPENSION NODE 266

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY $\omega = 2\pi f$	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION $A = \omega^2 D/386.4$ AT NODE				EFFECTIVENESS INDICATOR *
			15	31	63	95	15	31	63	95	
AWO, No Modification (Control)	12.052	75.72	0.380	0.362	0.320	0.245	5.64 g	5.37 g	4.75 g	3.64 g	0.05
Upper Floor Panel	12.187	76.57	0.330	0.323	0.291	0.239	5.01 g	4.9 g	4.42 g	3.63 g	0.06
Four Stanchions	12.041	75.66	0.349	0.335	0.313	0.244	5.17 g	4.96 g	4.64 g	3.61 g	0.05
Eight Stanchions	12.033	75.61	0.351	0.336	0.298	0.242	5.19 g	4.97 g	4.41 g	3.58 g	0.06
1/8" Shear Plate Above Door Opening	12.217	76.76	0.380	0.361	0.318	0.241	5.79 g	5.50 g	4.85 g	3.67 g	0.05
Boxed in Upper Floor Beams	12.078	75.89	0.338	0.322	0.289	0.235	5.04 g	4.80 g	4.31 g	3.50 g	0.06

* Effectiveness Indicator = $\left| \frac{g}{\sum \text{pseudoacceleration}} \right|$

The value closest to unity indicates the most effective modification.

TABLE 5.1-6 - UPPER FLOOR 3RD MODE PSEUDOACCELERATION WITH RESPECT TO SUSPENSION NODE 266

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY $\omega = 2\pi f$	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION $A = \omega^2 D/386.4$ AT NODE				EFFECTIVENESS INDICATOR *
			15	31	63	95	15	31	63	95	
AWO, No Modification (Control)	15.530	97.57	-0.470	-0.429	-0.318	-0.104	-11.58 g	-10.57 g	-7.83 g	-2.56 g	0.03
Upper Floor Panel	15.696	98.62	-0.358	-0.346	-0.266	-0.109	-9.01 g	-8.71 g	-6.69 g	-2.74 g	0.04
Four Stanchions	15.871	99.72	-0.074	-0.086	-0.225	-0.110	-1.90 g	-2.21 g	-5.79 g	-2.83 g	0.08
Eight Stanchions	15.963	100.30	-0.105	-0.096	-0.082	-0.094	-2.73 g	-2.50 g	-2.13 g	-2.45 g	0.10
1/8" Shear Plate Above Door Opening	15.530	97.57	-0.472	-0.431	-0.320	-0.105	-11.63 g	-10.62 g	-7.88 g	-2.59 g	0.03
Boxed in Upper Floor Beams	15.754	98.98	-0.316	-0.289	-0.229	-0.094	-8.01 g	-7.33 g	-5.81 g	-2.38 g	0.04

Effectiveness Indicator = $\left| \frac{g}{\sum \text{pseudoacceleration}} \right|$

The value closest to unity indicates the most effective modification.

TABLE 5.1-7 - LOWER FLOOR 2ND MODE PSEUDOACCELERATION WITH RESPECT TO SUSPENSION NODE 266

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY $\omega = 2\pi f$	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION $A = \omega^2 D/386.4$ AT NODE			
			1	17	49	81	1	17	49	81
AWO, No Modification (Control)	12.052	75.72	0.334	0.328	0.291	0.227	4.96 g	4.87 g	4.32 g	3.37 g
Upper Floor Panel	12.187	76.57	0.339	0.333	0.294	0.230	5.14 g	5.05 g	4.46 g	3.49 g
Four Stanchions	12.041	75.66	0.338	0.331	0.292	0.228	5.01 g	4.90 g	4.33 g	3.38 g
Eight Stanchions	12.033	75.61	0.338	0.332	0.294	0.228	5.0 g	4.91 g	4.35 g	3.37 g
1/8" Shear Plate Above Door Opening	12.217	76.76	0.333	0.326	0.288	0.225	5.08 g	4.97 g	4.39 g	3.43 g
Boxed in Upper Floor Beams	12.078	75.89	0.337	0.330	0.293	0.228	5.02 g	4.92 g	4.37 g	3.40 g

TABLE 5.1-8 - LOWER FLOOR 3RD MODE PSEUDOACCELERATION WITH RESPECT TO SUSPENSION NODE 266

MODEL DESCRIPTION	CARBODY BENDING FREQUENCY f (Hz)	CIRCULAR NATURAL FREQUENCY $\omega = 2\pi f$	RELATIVE VERTICAL NODAL DISPLACEMENT (D) AT NODE				PSEUDOACCELERATION $A = \omega^2 D/386.4$ AT NODE				
			1	17	49	81	1	17	49	81	
AWO, No Modification (Control)	15.530	97.57	-0.029	-0.029	-0.031	-0.032	-0.71 g	-0.71 g	-0.76 g	-0.79 g	
Upper Floor Panel	15.696	98.62	-0.053	-0.052	-0.051	-0.046	-1.33 g	-1.31 g	-1.28 g	-1.16 g	
Four Stanchions	15.871	99.72	-0.087	-0.083	-0.067	-0.049	-2.24 g	-2.14 g	-1.72 g	-1.26 g	
Eight Stanchions	15.963	100.30	-0.096	-0.093	-0.079	-0.056	-2.50 g	-2.42 g	-2.06 g	-1.46 g	
1/8" Shear Plate Above Door Opening	15.530	97.57	-0.031	-0.031	-0.033	-0.033	-0.76 g	-0.76 g	-0.81 g	-0.81 g	
Boxed in Upper Floor Beams	15.754	98.98	-0.051	-0.050	-0.049	-0.044	-1.29 g	-1.27 g	-1.24 g	-1.12 g	

5.3.1 Four Stanchion Arrangement

The first case run was for four stanchions symmetrically located about the car centre at BL 14.0 and stations 477.5 and 542.5. Since the model represents only 1/4 of the car only one stanchion of the four was added to the model. Results for this mod are available in Tables 5.1-1 through 5.1-8 under the label "four stanchions". The mode shapes in Figure 31 and 32 show that little change occurred to the 12 Hz bending mode as expected but that the upper deck deflection of the 15.9 Hz mode has been zeroed in the vicinity of the stanchion. Elsewhere the deflection has been reduced. The maximum deflection now occurs further towards the car end at station 412.5. This mod is the second most effective of the five mods.

5.3.2 Eight Stanchion Arrangement

The need to move fully constrain the upper deck led to the eight stanchion arrangement being run on the model. Only two stanchions were modelled in the ¼ car model representing eight stanchions at BL ± 14.0 and stations 412.5, 477.5, 542.5 and 607.5

The predicted modes and responses are contained in Tables 5.1-1 through 5.1-8 under the label "eight stanchions". In figures 33 and 34 are the mode shapes for the 12 Hz and 16 Hz modes. While little effect is seen of the stanchions in the 12 Hz mode, the upper deck heave motion is virtually eliminated in the 16 Hz mode making the upper and lower decks similar. This mod is the most effective of the structural mods described here based upon the modal changes. The test results discussed in Volume II show that the stanchions were effective at controlling the 16 Hz mode only.

5.4 STIFFENED FLOOR BEAMS

Another of the early proposed solutions to the dither problem was to stiffen the upper deck cross beams. This was accomplished by boxing in the tapered beams with the addition of web plates connecting the upper and lower cord members the first three beams either side of the car centre. The results are tabulated in Tables 5.1-1 through 5.1-8 under the label "boxed in Upper Floor Beams". The mode shapes are shown in Figures 35 and 36. This mod is not too effective and ranks equal with the stiffened floor panels.

5.5 INCORPORATION OF STIFFENED DOOR PLATES

In most rail cars the greatest vertical deflections occur at door openings where the high shear forces must be reacted by a small number of beam elements in bending. By strengthening the shear capability of the structure around the doors, the body bending frequencies can be raised to remove troublesome resonances. Such was the case with the door plates where a 1/8" aluminum plate was placed above each door between STA 214.62 and 347.5 and WL 106.68 to 137.6. The predicted modal behaviour is shown in Tables 5.1-1 through 5.1-8 under the label "1/8" Shear Plate above Door Opening. The corresponding mode shape plots at 12.2 Hz and 15.5 Hz are shown in Figures 37 and 38. As modelled, this change did not make a significant difference to the car behaviour. This is likely because either the plate was not thick enough or did not cover enough area. Practical considerations may rule out large changes to the assumed plate, however.

6.0 CONCLUSION

6.1 DYNAMIC MODEL

A series of nine vertical dynamic models have been developed to represent the Bi-level I, II and III cars. Each was run on UTDC in-house software to extract eigenvalues (natural frequencies and modes) and to predict the response to random vertical and cross-level track input. The results of the modelling are:

1) no one modification studied was capable of reducing dither vibration to an acceptable level (< half of initial value).

2) The ranking in order of effectiveness of proposed mods is:

   a) stiffen carbody bending stiffness;
   b) use secondary damper bush stiffness mod;
   c) add truck primary dampers; and
   d) stiffen upper deck beams and floor or add stanchions.

6.2 FINITE ELEMENT MODEL

A 1/4 car ANSYS based finite element model was derived and used to assess the effect upon modal frequencies and relative acceleration (pseudo acceleration) of various modifications to the structure. The results of this study are:

1) Two modes primarily affect the dither vibration; a 12 Hz body bending mode causing both decks to vibrate equally and a 15-16 Hz mode associated with heave motion of the central portion of the upper deck;

2) The ranking of modifications studied in order of effectiveness from Section 3.2 of (7) is:

   a) eight stanchions installed between decks;
   b) four stanchions installed between decks;
   c) boxed in upper floor beams;
   d) upper floor panel installed; and
   e) 1/8" shear plate above door.

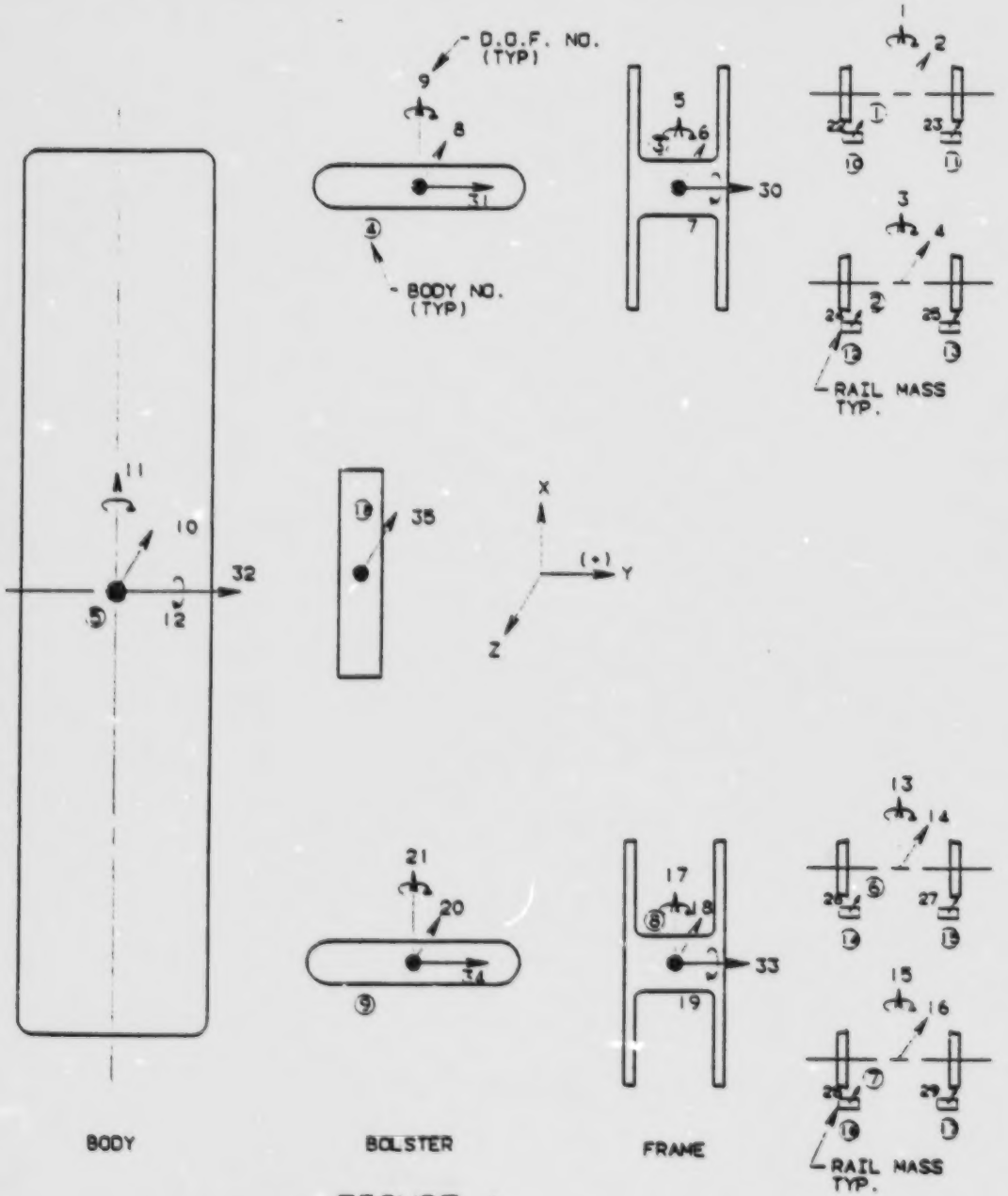

D.O.F. NO. (TYP)
BODY NO. (TYP)
RAIL MASS TYP.
RAIL MASS TYP.
BODY
BOLSTER
FRAME

FIGURE 1

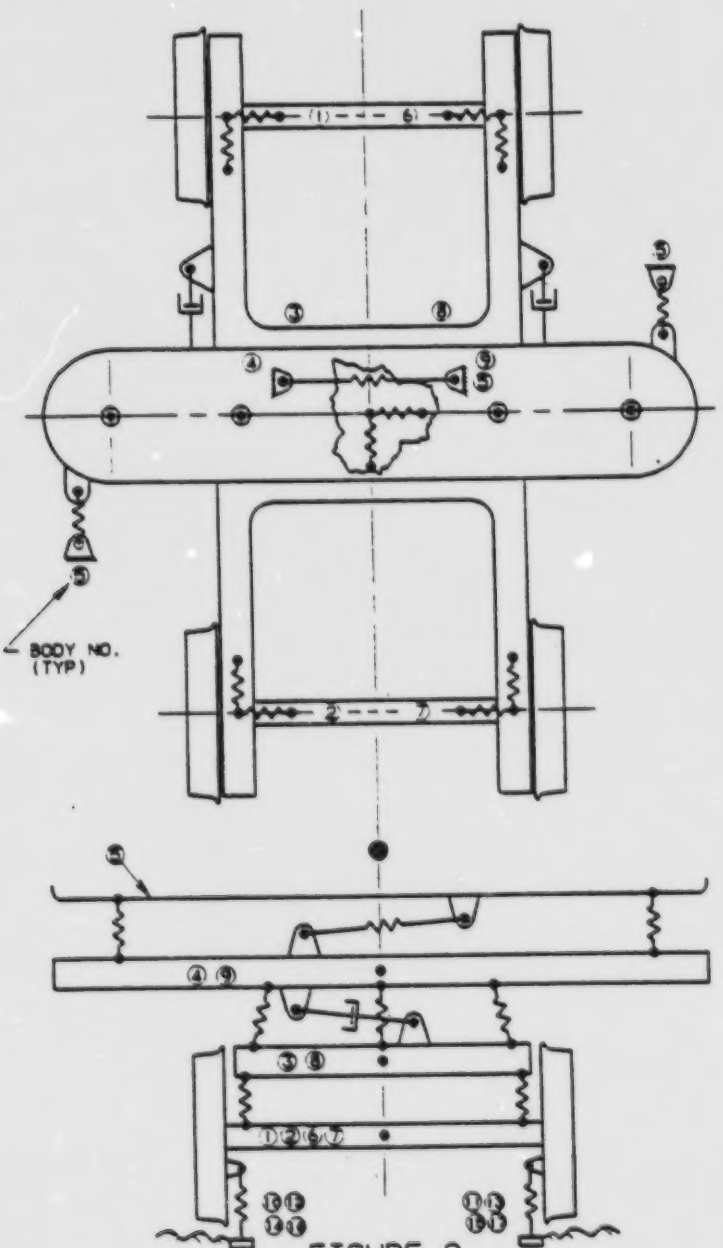


FIGURE 2

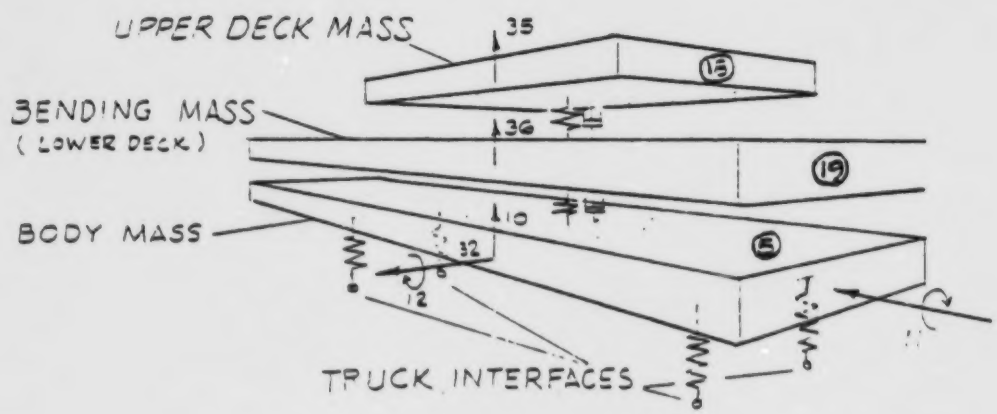
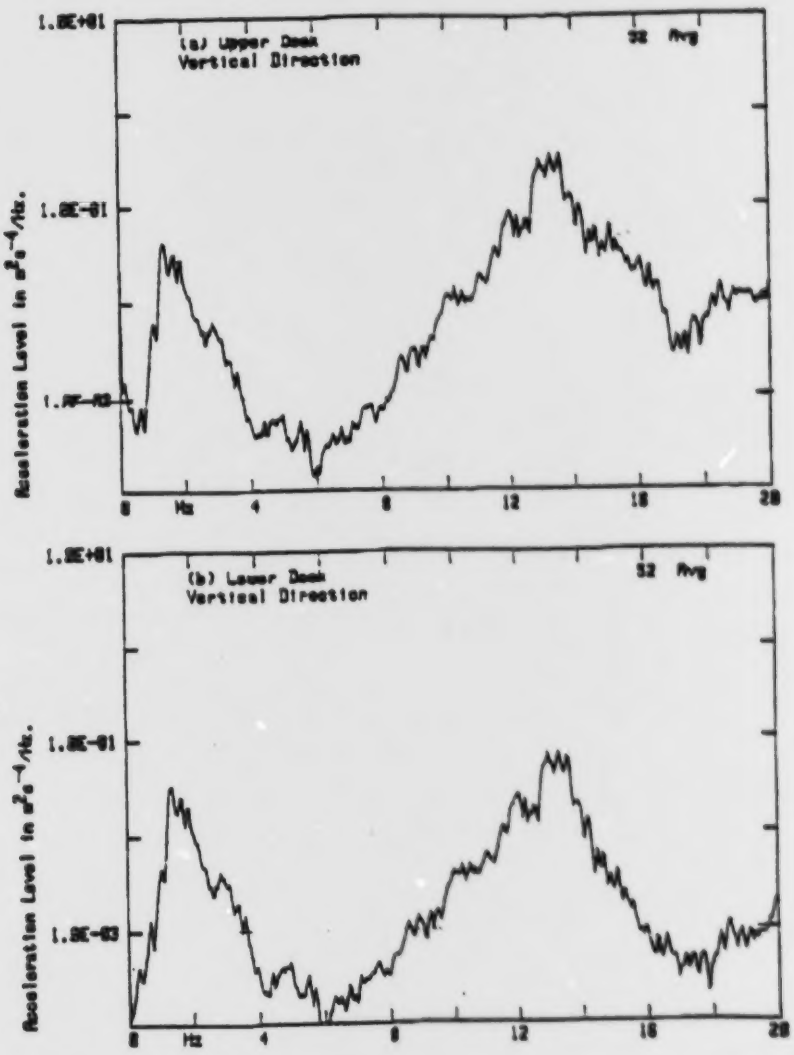


FIGURE 3 : 3 MASS BODY MODEL

NOTE: NUMBERS REFER TO DEGREES OF FREEDOM ASSIGNED; ARROW INDICATES DIRECTION CIRCLED NUMBERS ARE MASSES



COp0001, C015d02 Coach 2114, Port Credit to Oakville,~1R min
MTC (416) 235-3957 1987.05.08 GO bilevel testing 1987.04.13

FIGURE 4: VERTICAL ACCELERATION SPECTRA - CAR 2144

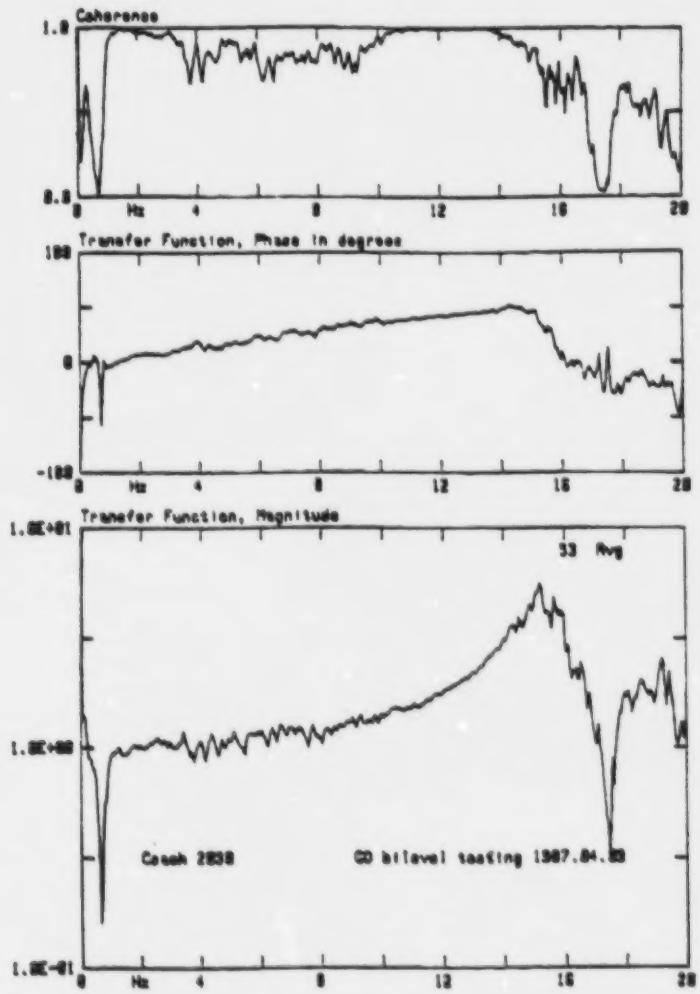


FIGURE 5: TYPICAL TRANSFER FUNCTION - CAR 2030

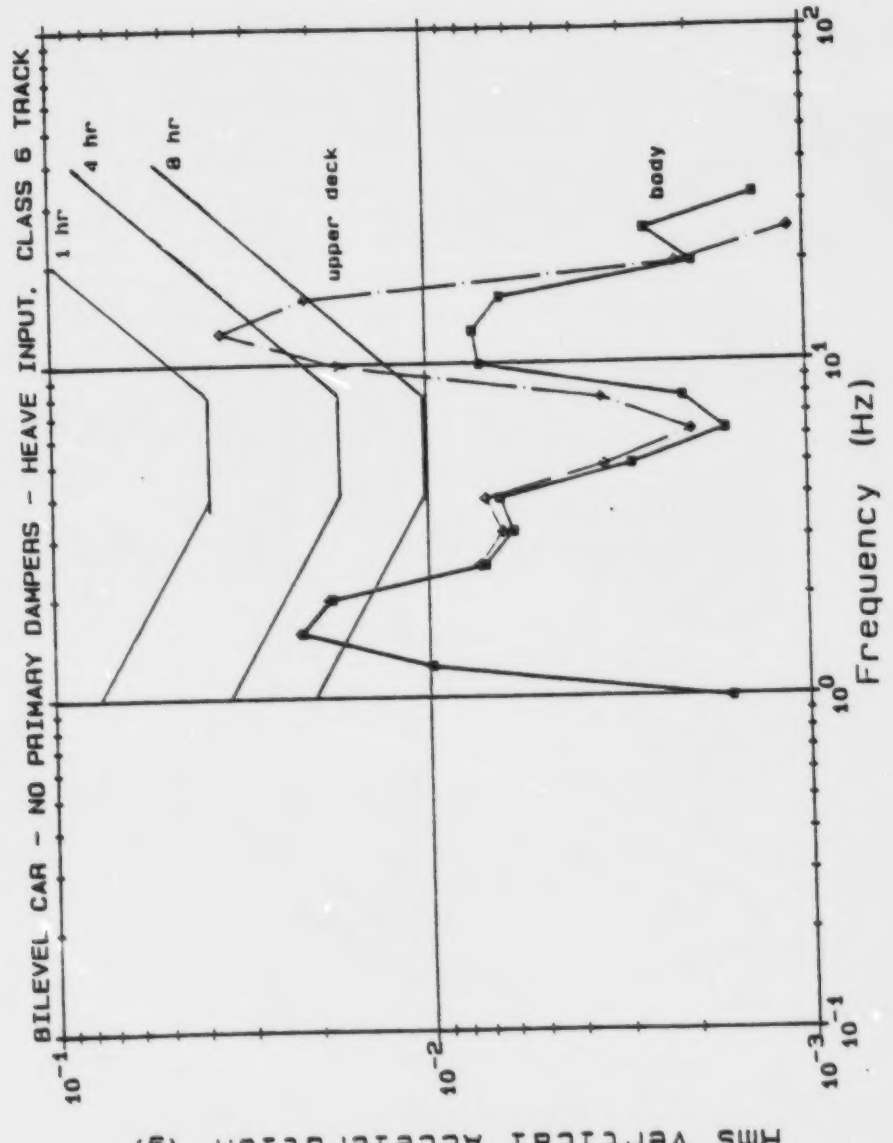


FIGURE 6: CALC'D RESPONSE ON CLASS 6 TRACK 80 MPH (HEAVE INPUT) - 2 MASS BODY

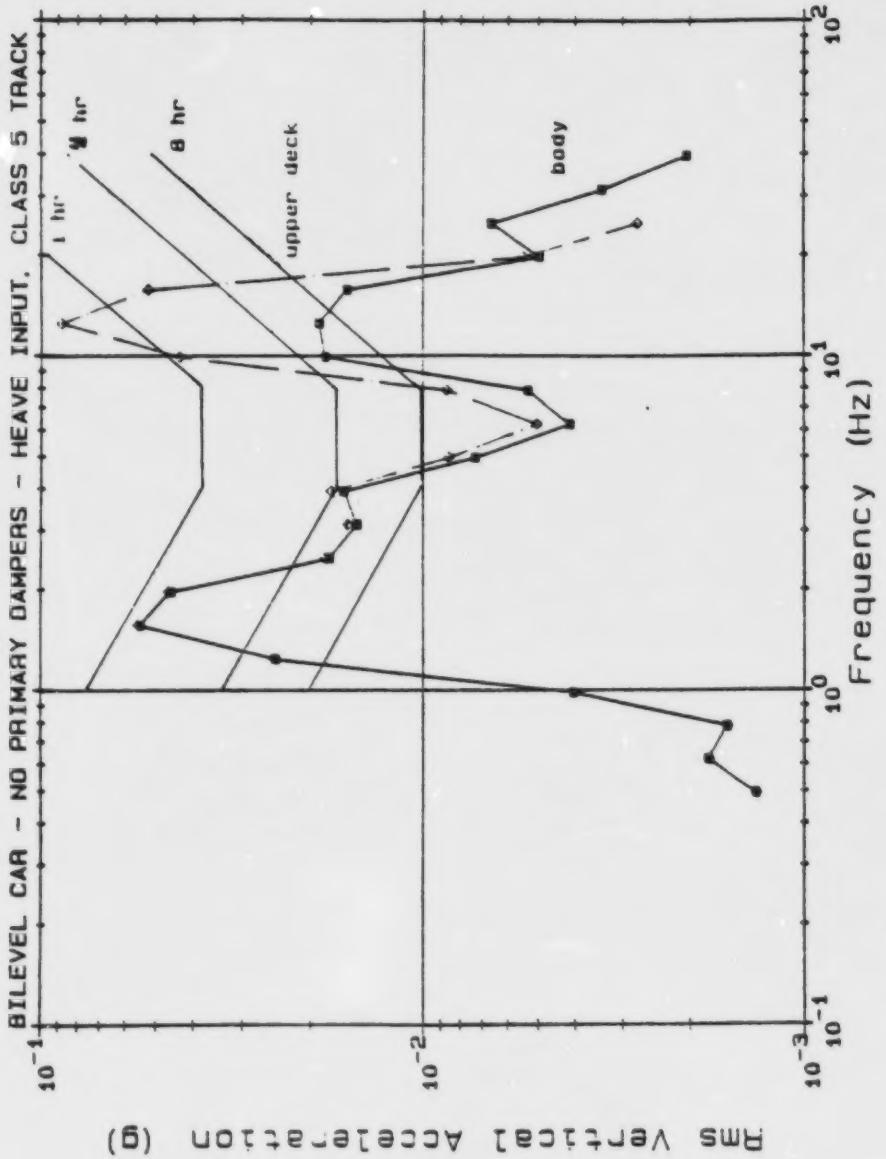


FIGURE 7: CALC'D RESPONSE ON CLASS 6 TRACK 80 MPH
(HEAVE INPUT) - TRUCK

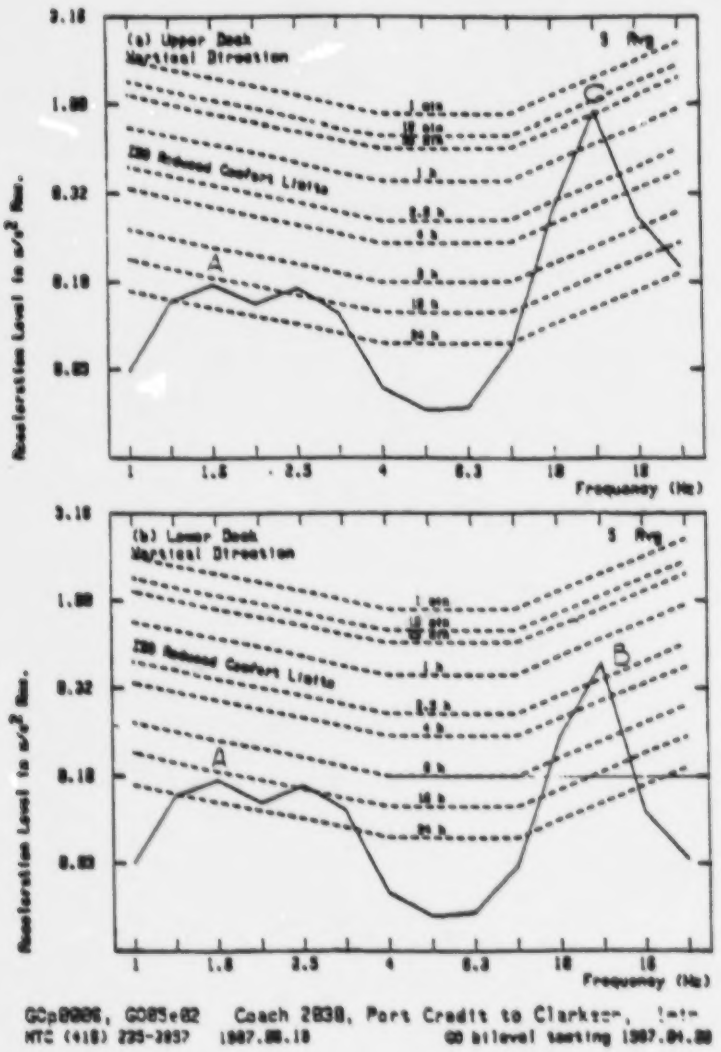


FIGURE 8: TYPICAL MEASURED RIDE RESPONSE
(REF. 3, MTC) CAR 2030

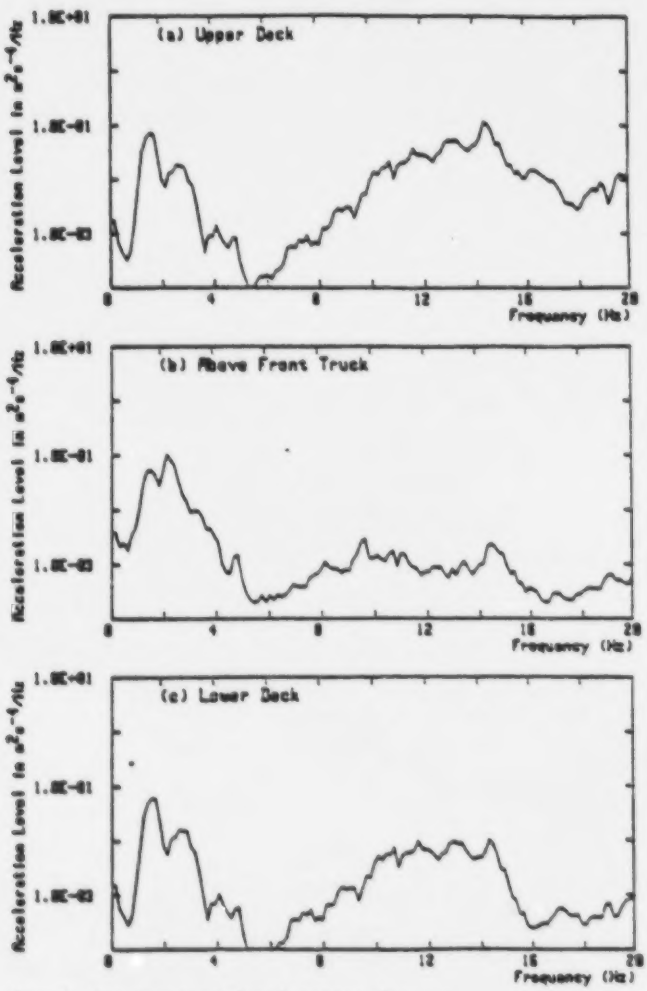


Figure 38/ Vertical Acceleration Spec a
Bi-Leve Car Fair Tangent

FIGURE 9: VERTICAL ACCELERATION SPECTRA FROM (REF. 4, P. 60)

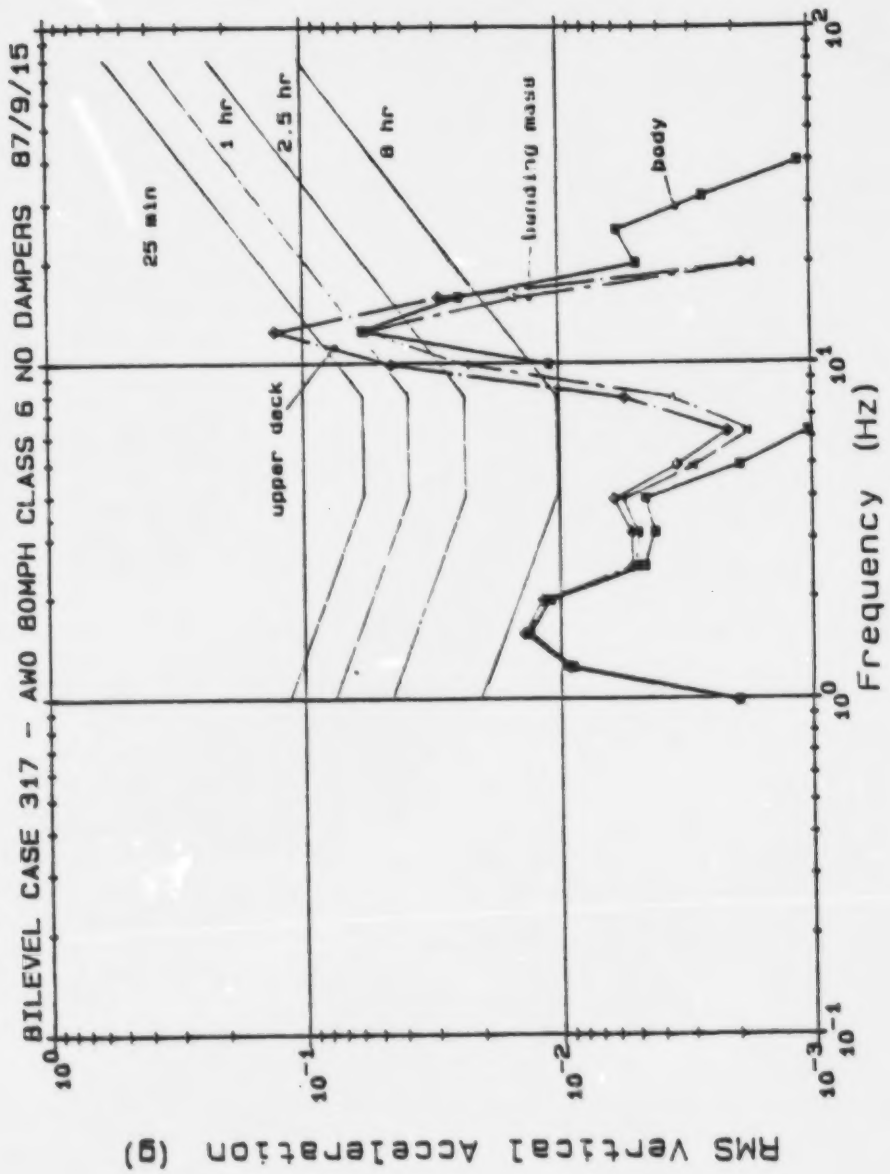


FIGURE 10: CALC'D RIDE - NO DAMPERS -
3 MASS BODY MODEL HEAVE INPUT

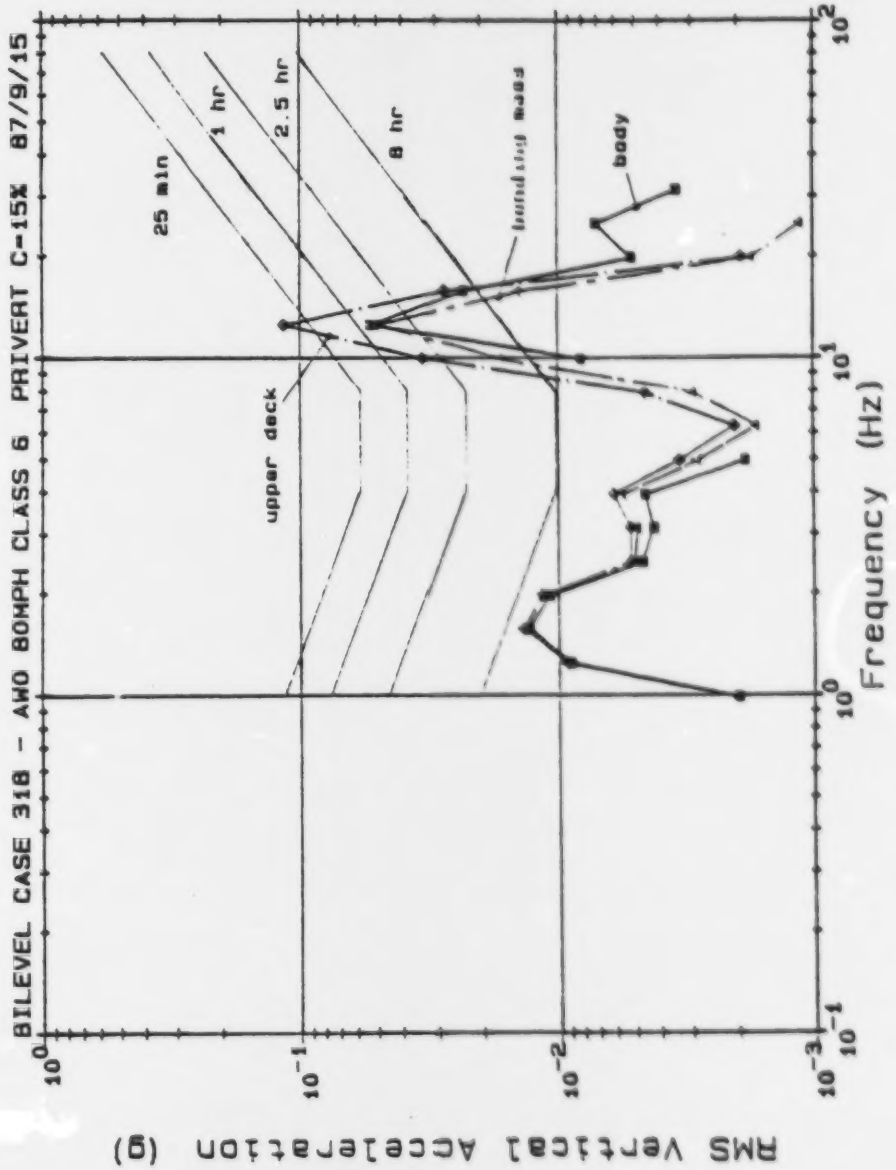


FIGURE 11: CALC'D RIDE - PRIMARY TRUCK DAMPERS AT 15% CRITICAL - 3 MASS BODY MODEL

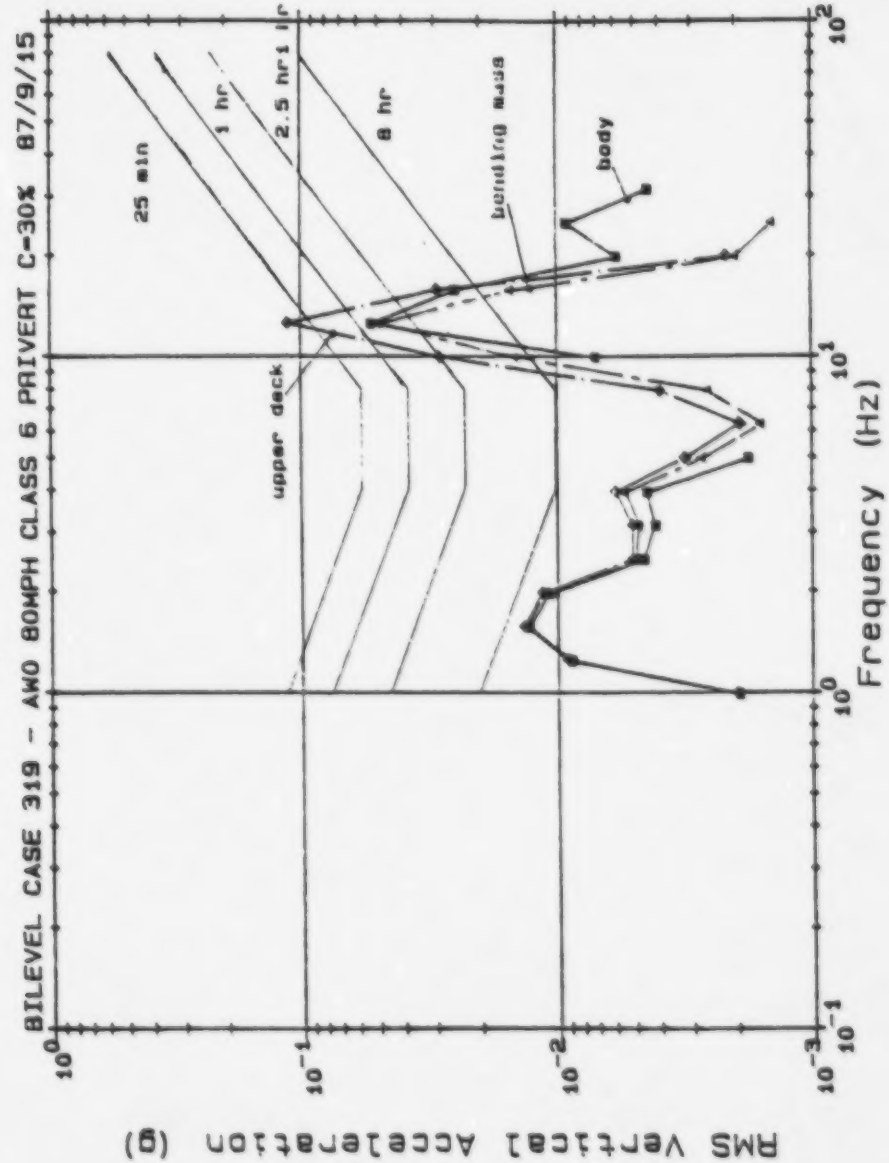


FIGURE 12: CALC'D RIDE - PRIMARY TRUCK DAMPERS AT 30% CRITICAL - 3 MASS BODY MODEL

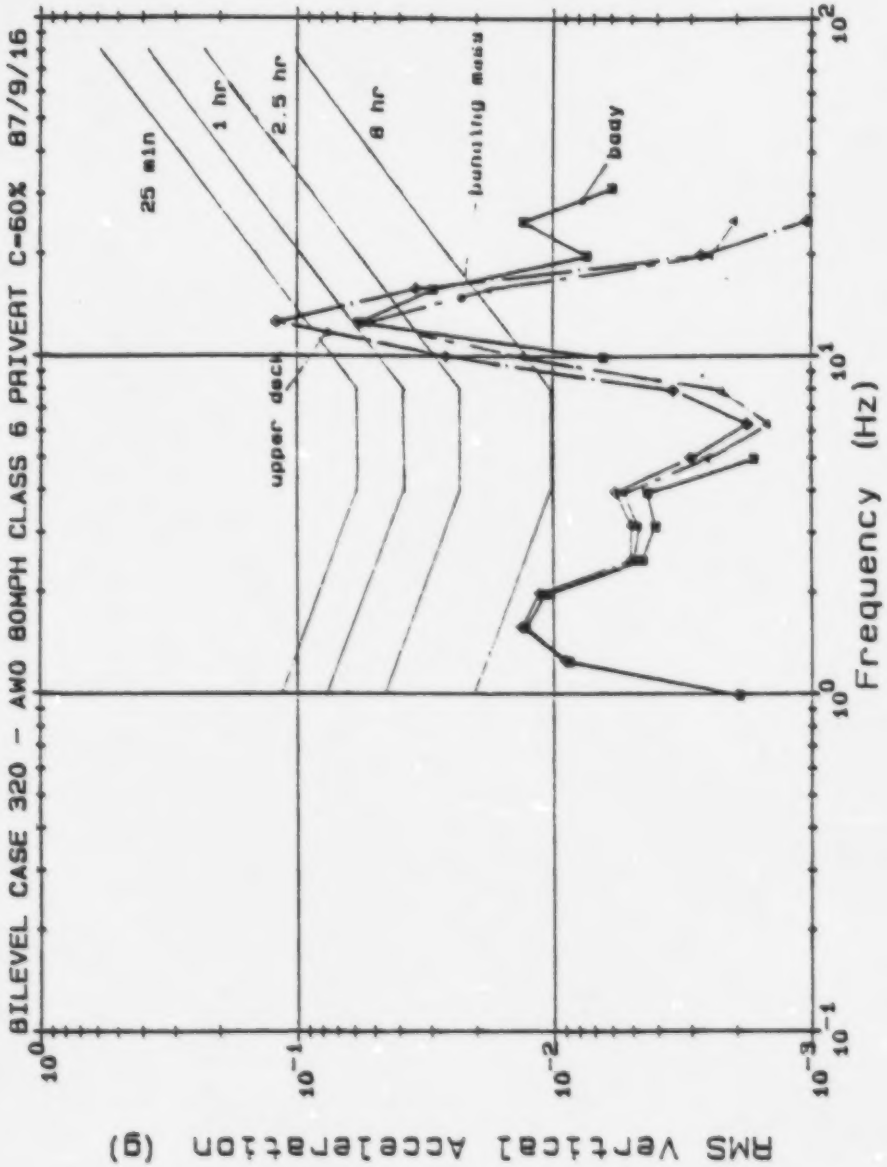


FIGURE 13: CALC'D RIDE - PRIMARY TRUCK DAMPERS AT 60% CRITICAL - 3 MASS BODY MODEL

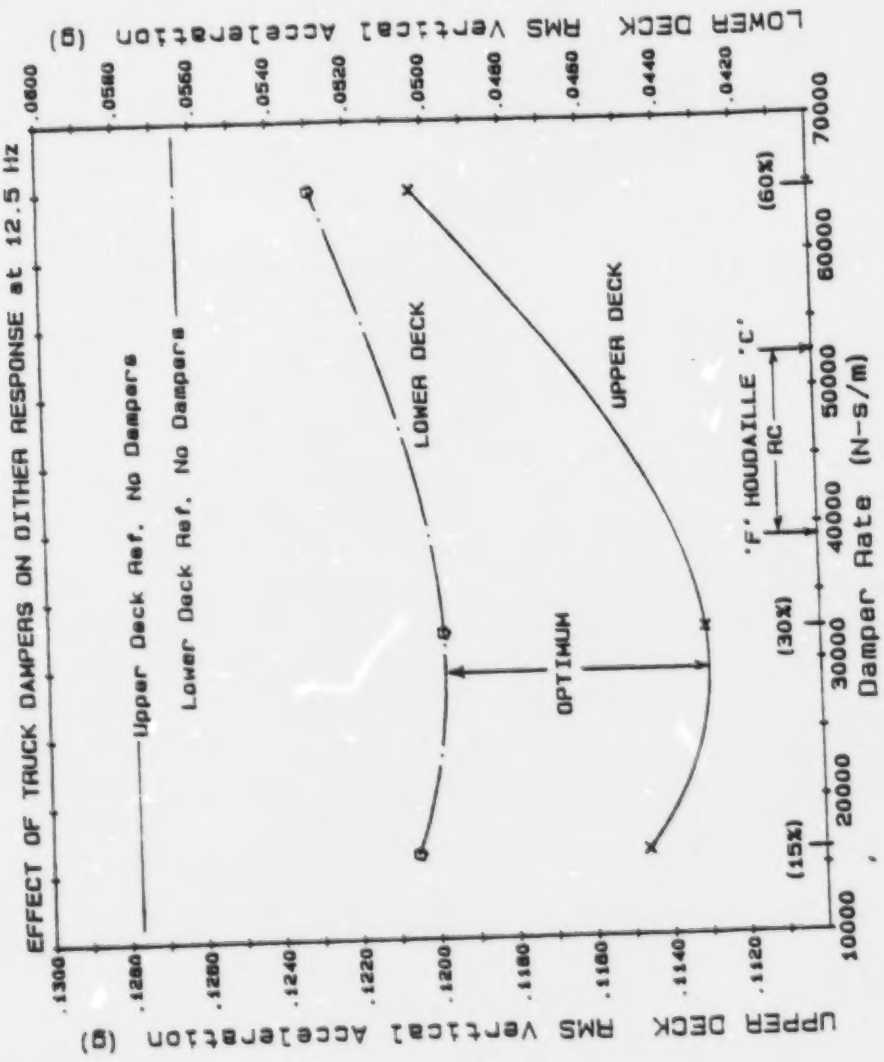


FIGURE 14: EFFECT OF PRIMARY TRUCK DAMPERS ON DITHER RESPONSE

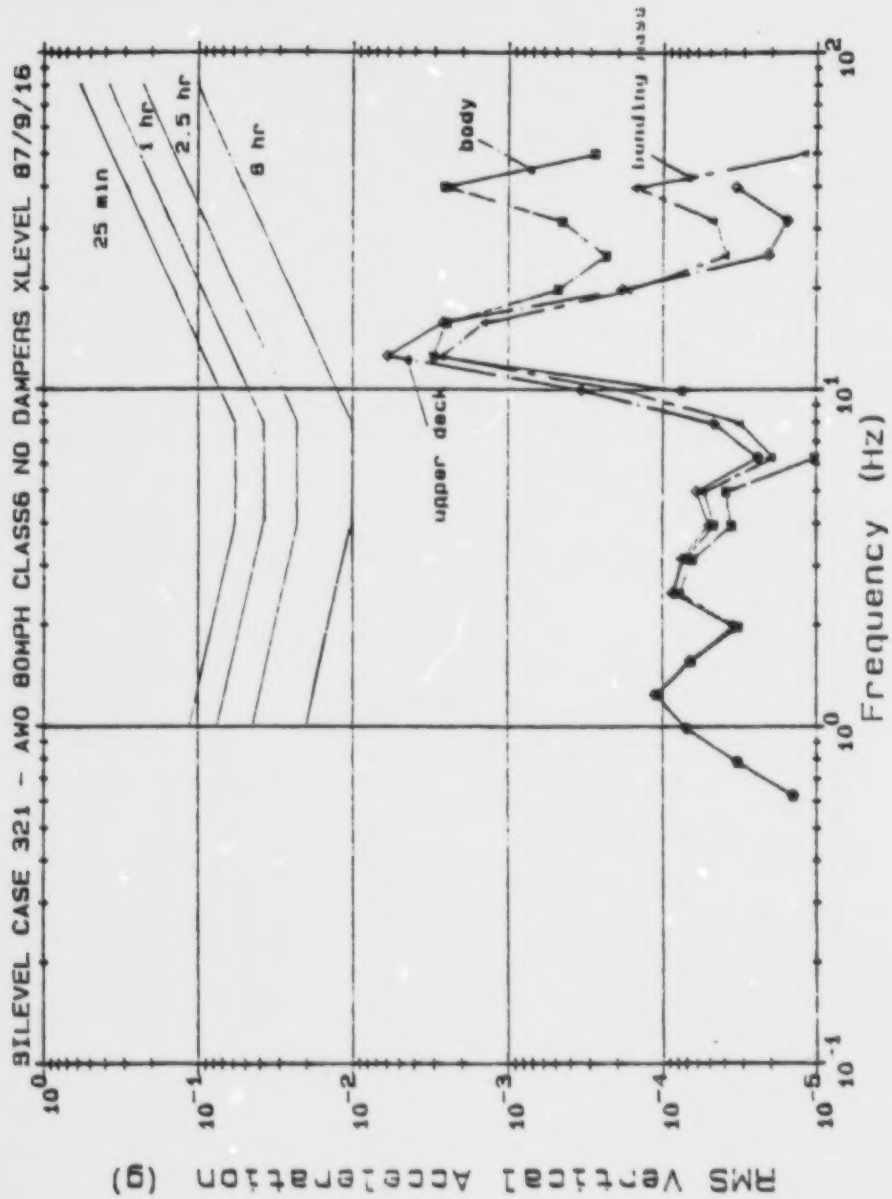


FIGURE 15: CALC'D RIDE - NO DAMPERS - 3 MASS BODY MODEL - X-LEVEL INPUT

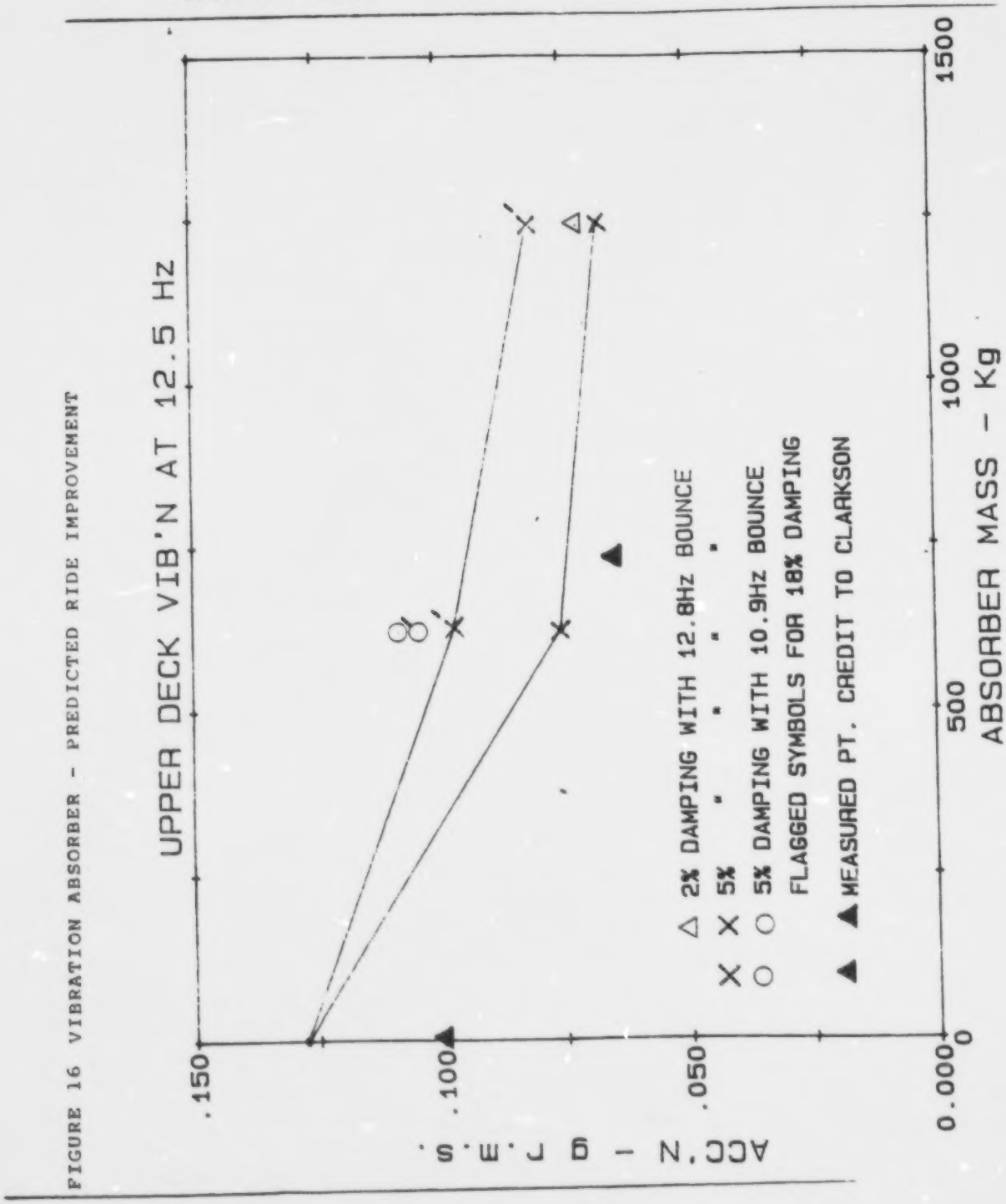


FIGURE 16 VIBRATION ABSORBER - PREDICTED RIDE IMPROVEMENT

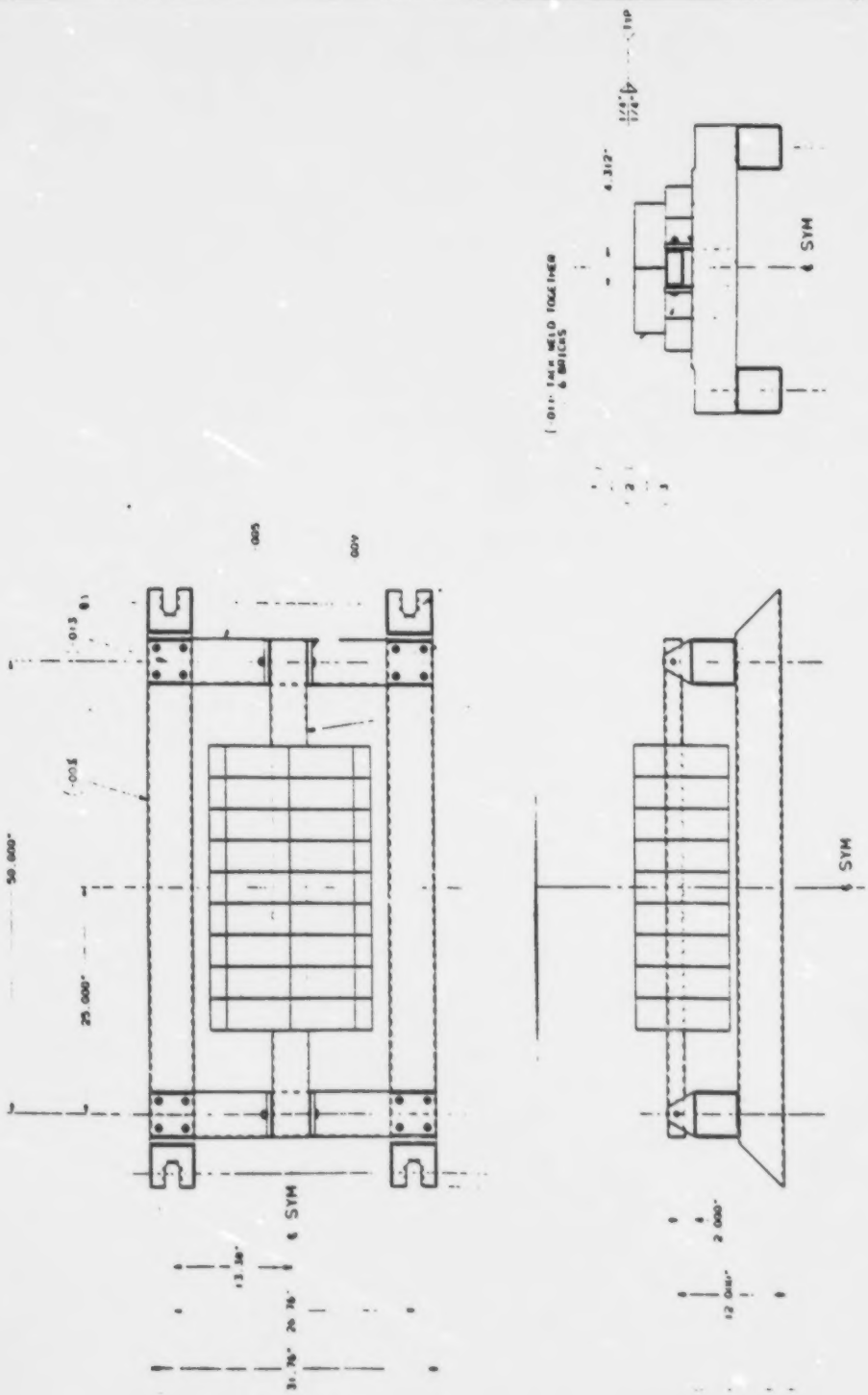


FIGURE 17 VIBRATION ABSORBER - GENERAL ASSEMBLY

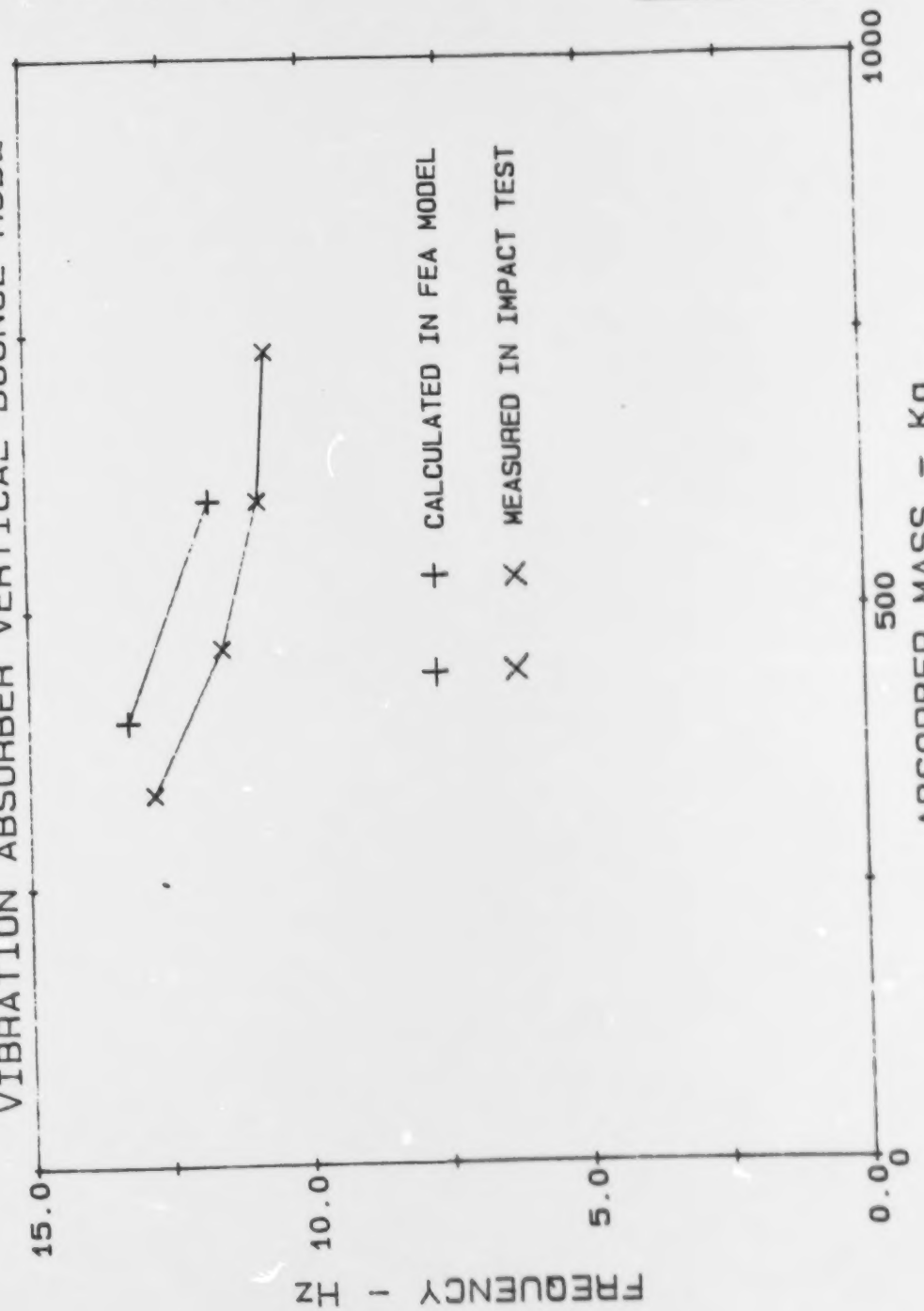


FIGURE 18 VIBRATION ABSORBER FREQUENCY VERSUS APPLIED MASS

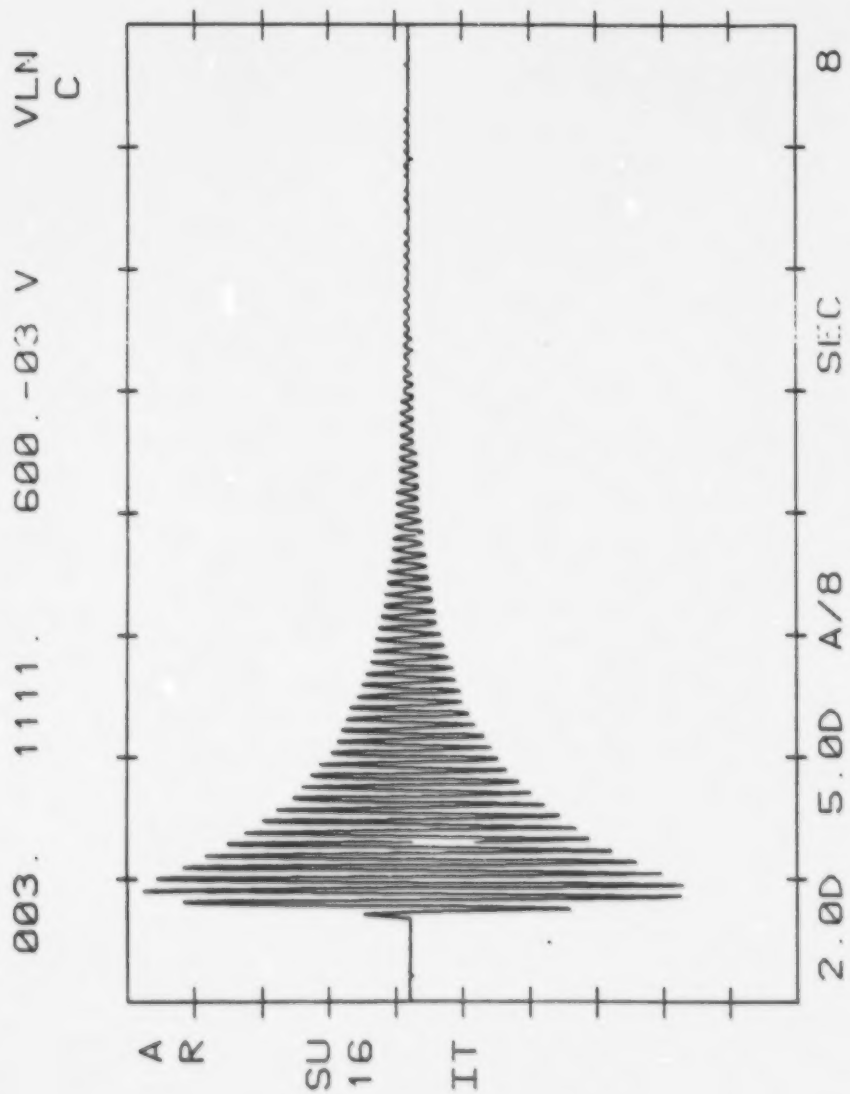


FIGURE 19 VIBRATION ABSORBER - IMPACT TEST DECAY PLOT AT CENTRE OF BEAM

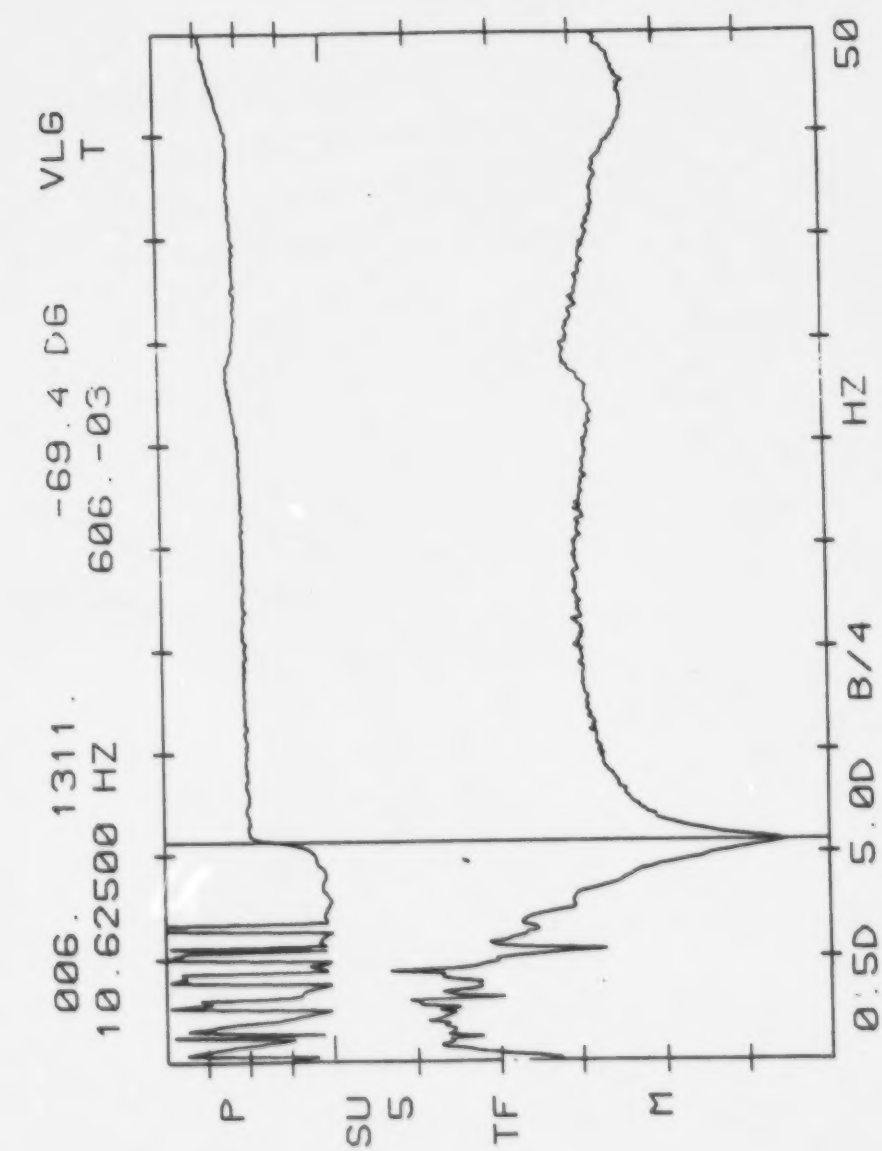


FIGURE 20 VIBRATION ABSORBER - MEASURED IMPEDANCE AT CENTRE OF BEAM FOR FOR 11 WEIGHTS APPLIED

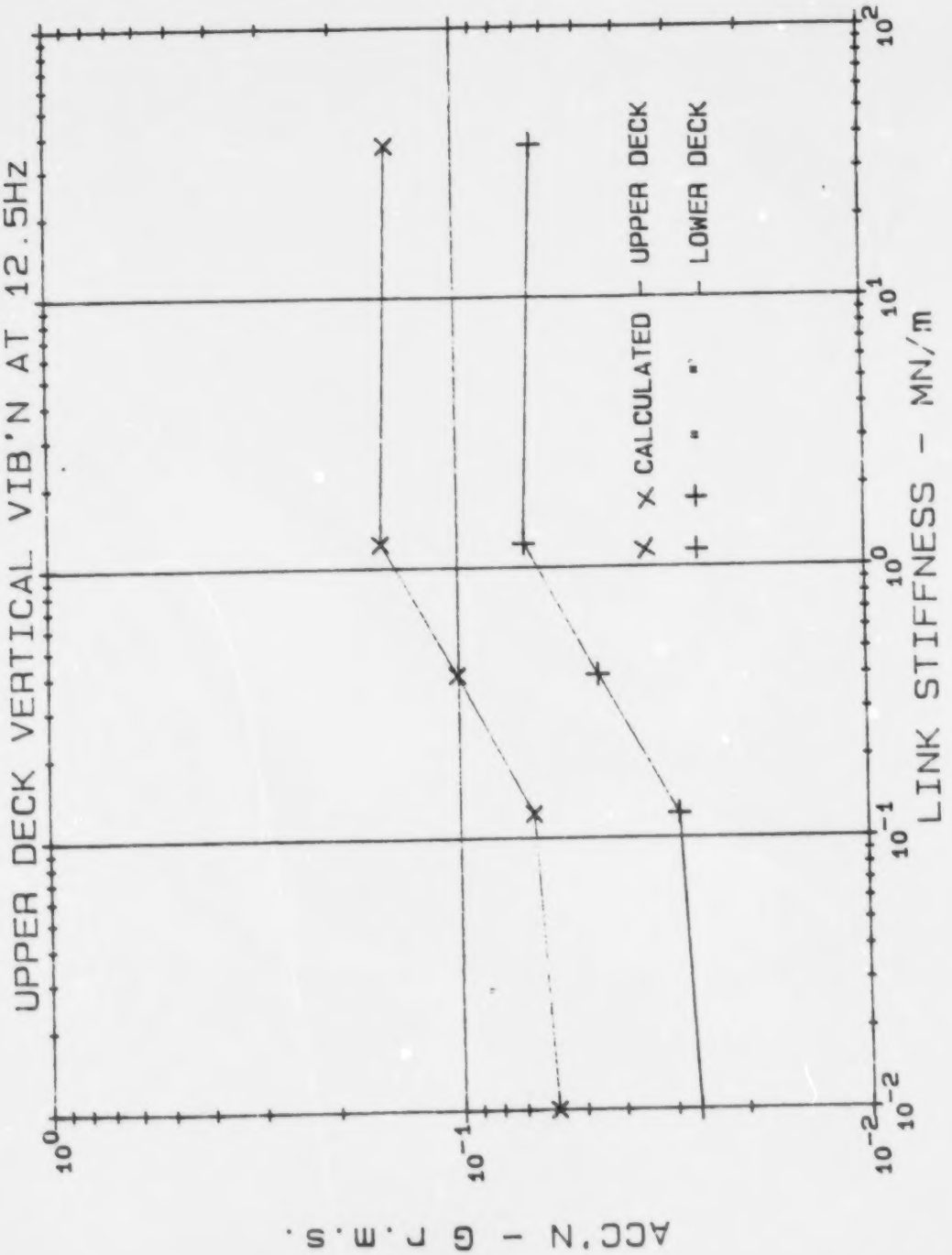


FIGURE 21 SECONDARY DAMPER BUSH MOD: PREDICTED VERTICAL RIDE

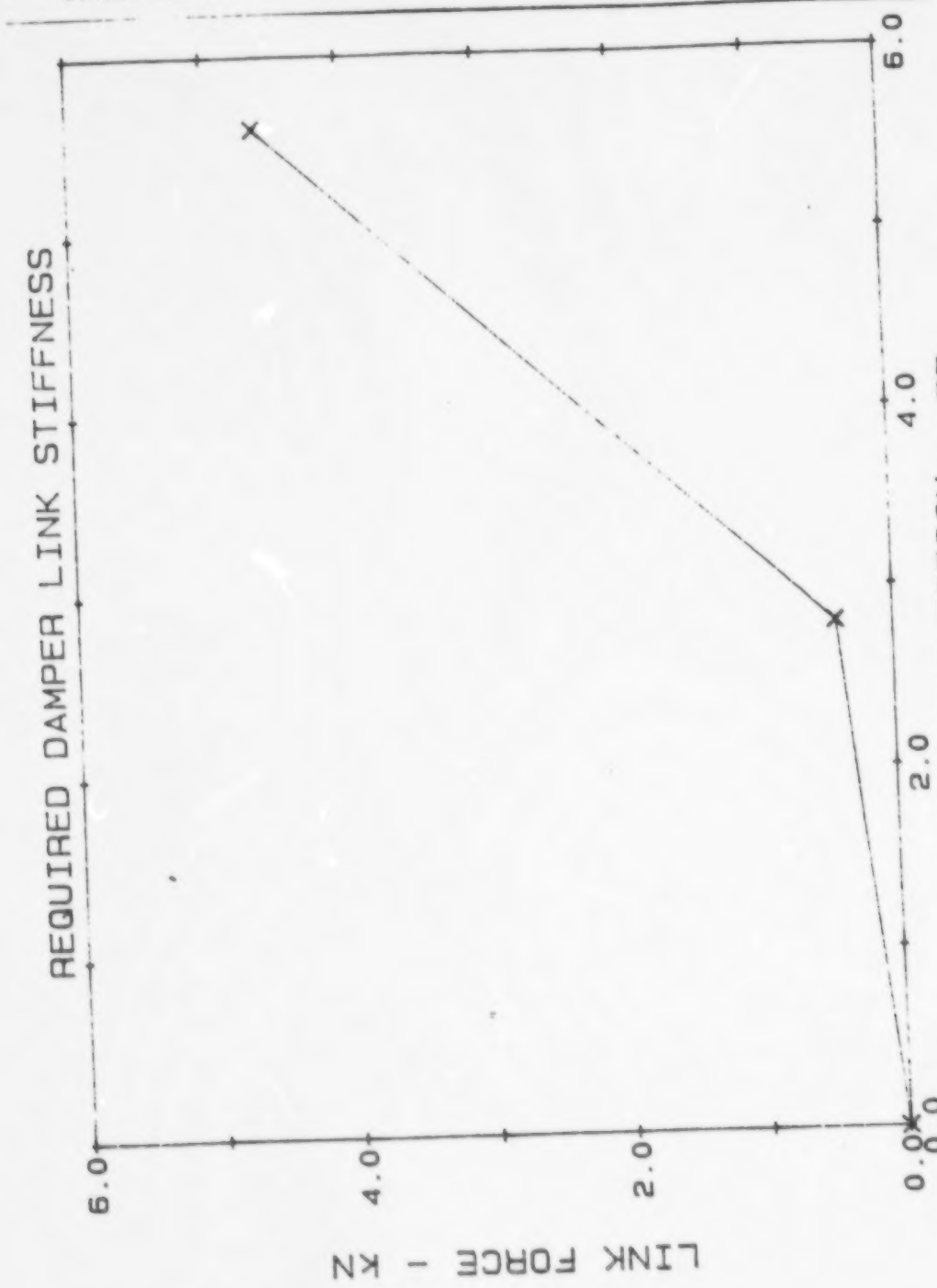


FIGURE 22 SECONDARY DAMPER BUSH MOD: LINK STIFFNESS REQUIRED FOR OPTIMUM RIDE

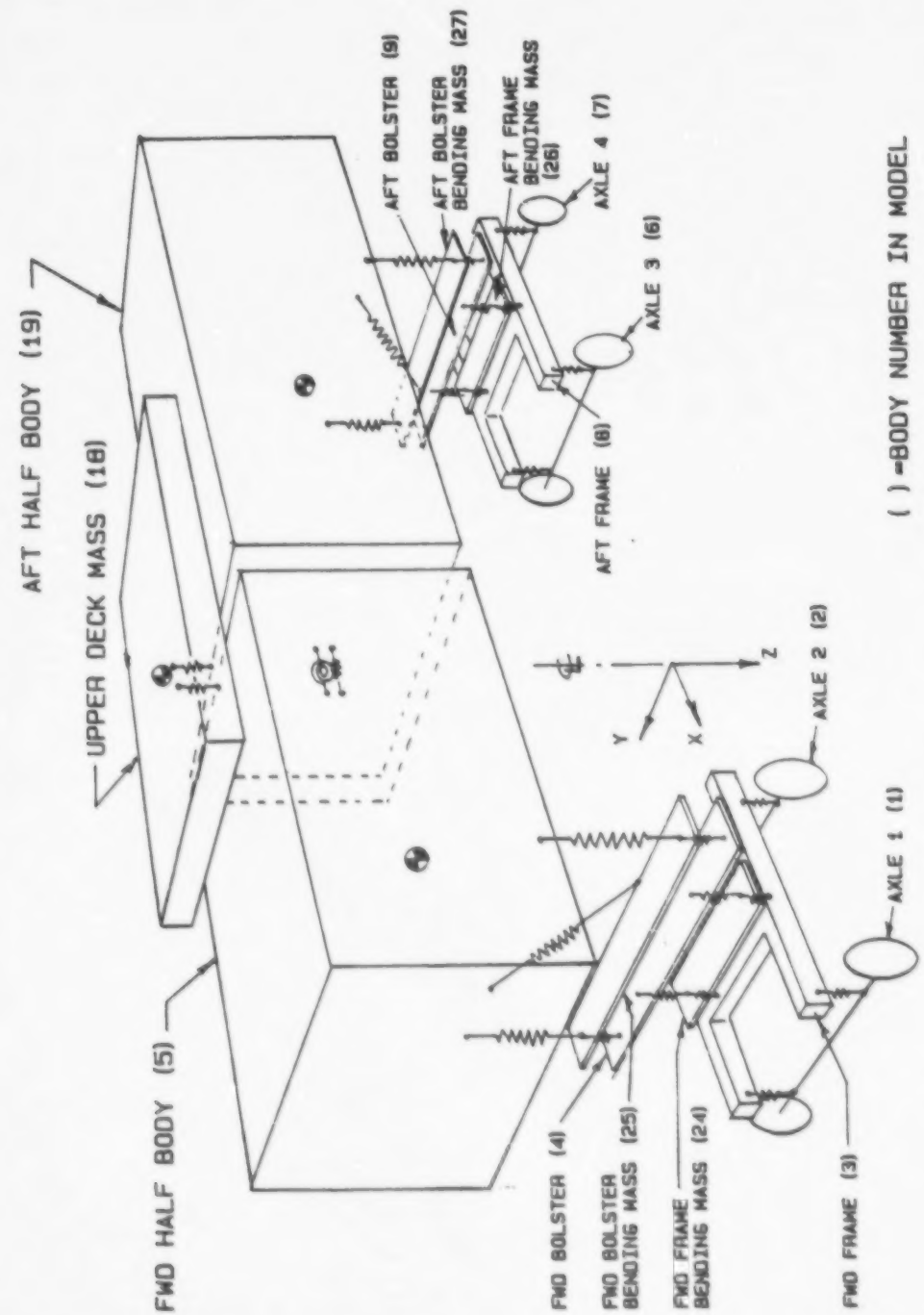


FIGURE 23 BILEVEL8 MODEL

FIGURE 23

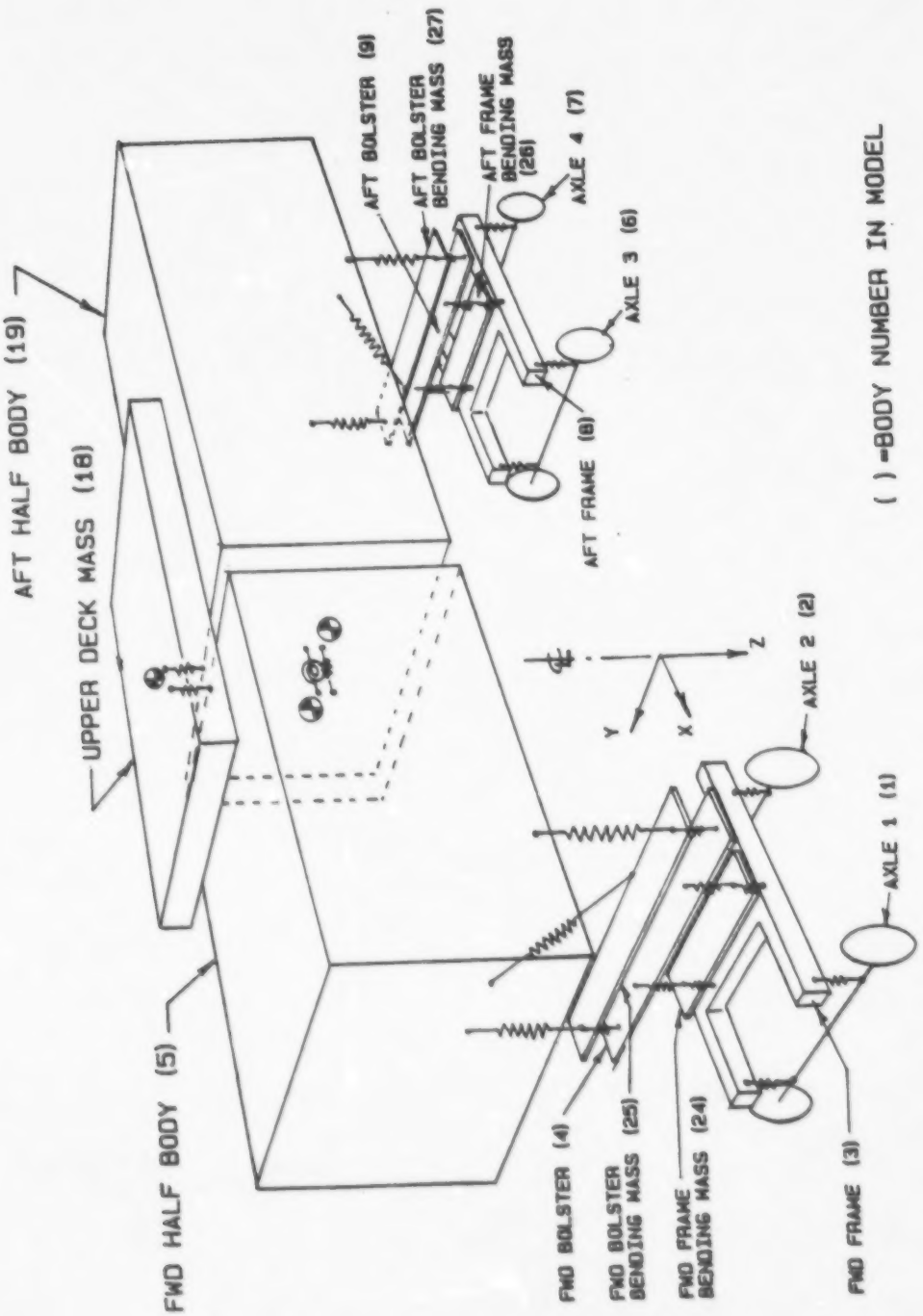


FIGURE 24 BILEVEL9 MODEL

FIGURE 24

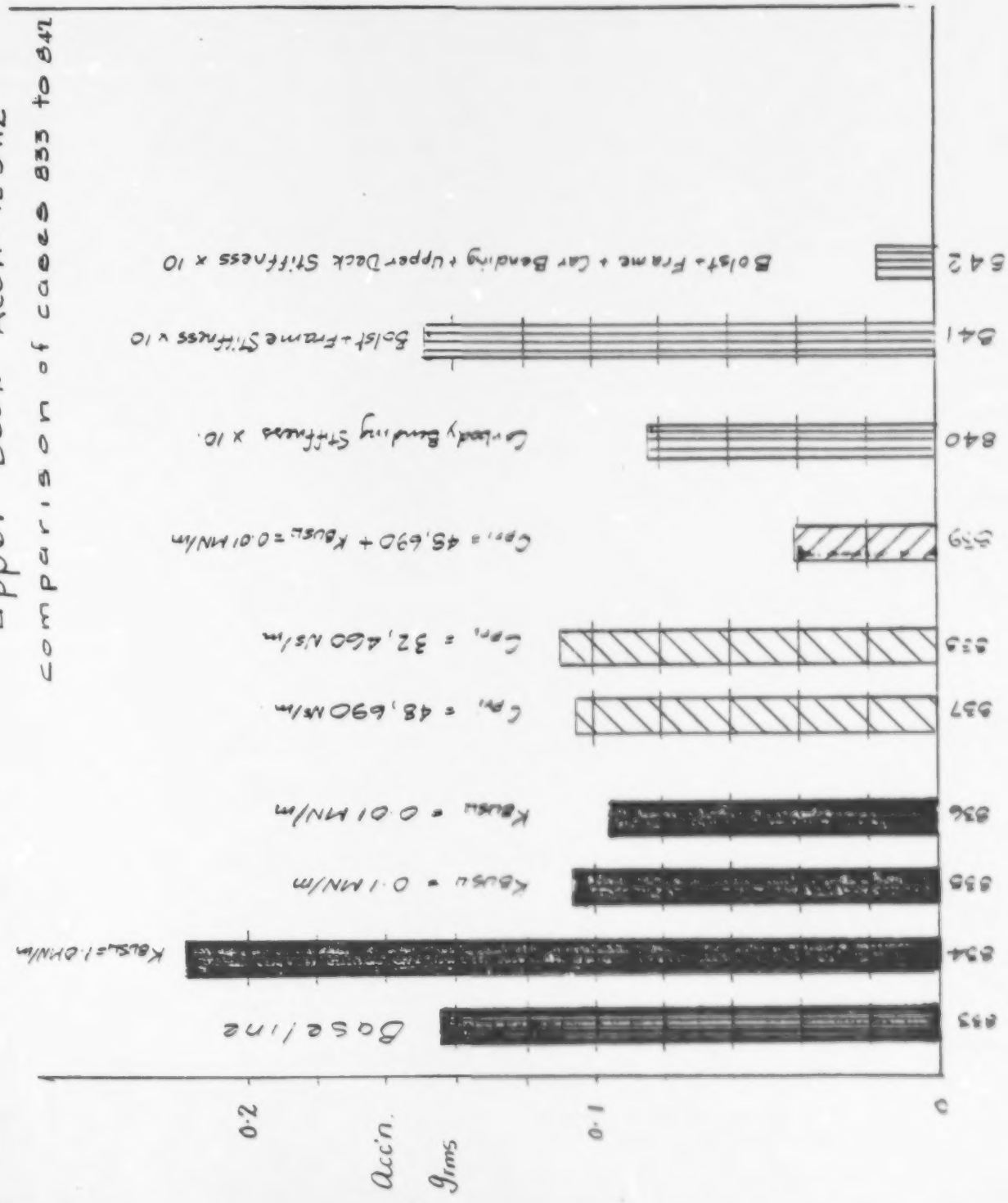

Bi-Level Car Model
Upper Deck Acc'n - 12.5 Hz
Comparison of cases 833 to 842
833
834
835
836
837
839
839
840
841
842
Base line
$K_{BUSH} = 1.0 MN/m$
$K_{BUSH} = 0.1 MN/m$
$K_{BUSH} = 0.01 MN/m$
$C_{PRI} = 48,690 Ns/m$
$C_{PRI} = 32,460 Ns/m$
$C_{PRI} = 45,690 + K_{BUSH} = 0.01 MN/m$
Car body Bending Stiffness x 10
Bolst + Frame Stiffness x 10
Bolst + Frame + Car Bending + Upper Deck Stiffness x 10
0
0.1
0.2
Acc'n
g rms

FIGURE 25

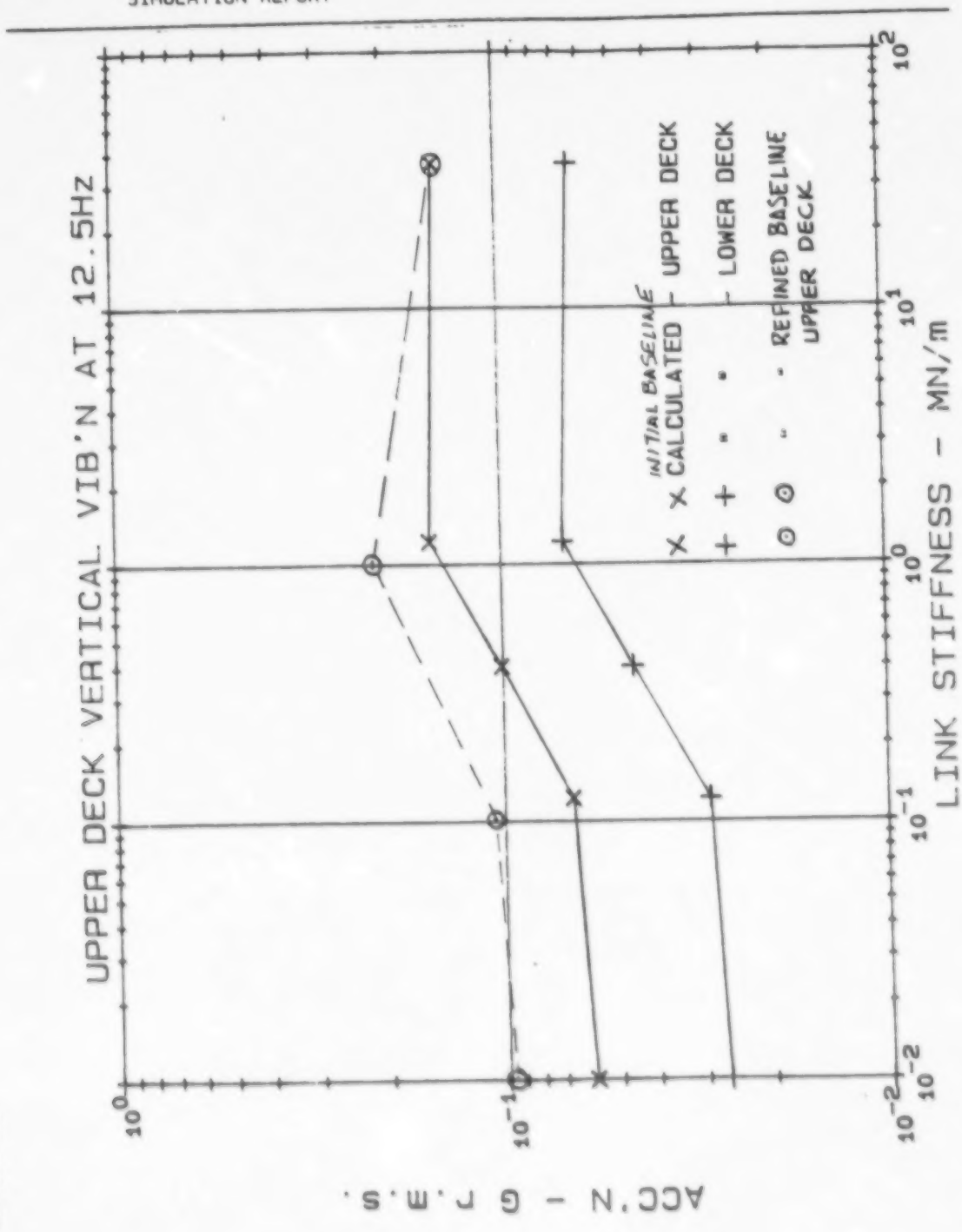

UPPER DECK VERTICAL VIB'N AT 12.5Hz
ACC'N - G's
LINK STIFFNESS - MN/m
INITIAL BASELINE
CALCULATED
UPPER DECK
LOWER DECK
REFINED BASELINE
UPPER DECK
10^{0}
10^{-1}
10^{-2}
10^{2}
10^{1}
10^{0}
10^{-1}
10^{-2}

FIGURE 26

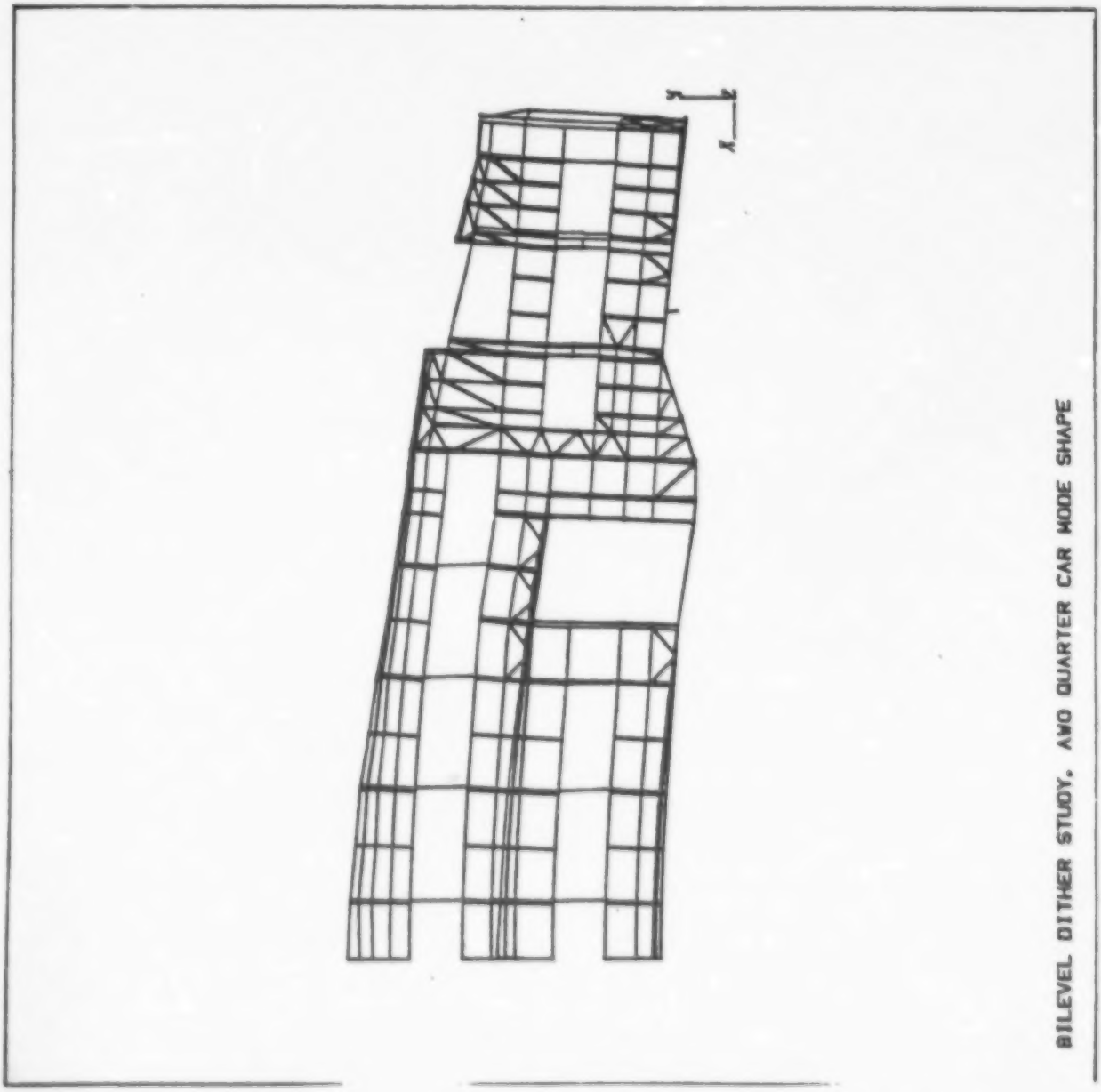


FIGURE 27 BASELINE CAR - FIRST BENDING MODE SHAPE

BILEVEL DITHER STUDY, AVO QUARTER CAR MODE SHAPE

ANSYS 4.2
NOV 5 1987
12:34:41
DISPL.
STEP=1
ITER=3
FREQ=15.5
DMX=1.0
ZV=-1
DIST=320
XF=285
YF=105
ZF=28

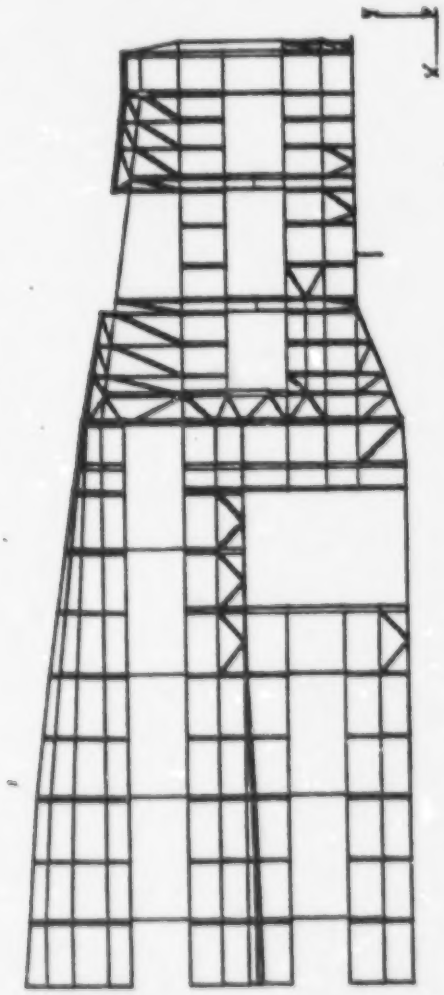


FIGURE 28 BASELINE CAR - UPPER DECK HEAVE MODE SHAPE

BI-LEVEL DITHER STUDY, QUARTER CAR DYNAMIC MODEL

ANSYS 4.2
NOV 10 1987
11:48:10
DISPL.
STEP=1
ITER=2
FREQ=12.2
ZV=-1
DIST=280
XF=205
YF=105
ZF=28

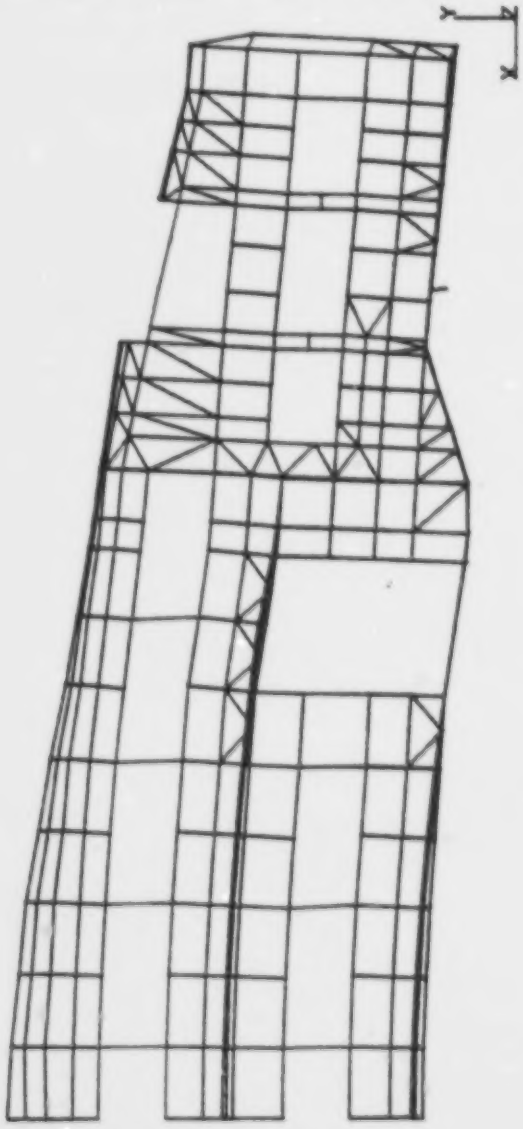


FIGURE 29 FLOOR PANEL MOD - FIRST BENDING MODE SHAPE

BI-LEVEL DI[illegible] STUDY. QUARTER CAR MODEL W/PROPOSED FL. PNL.

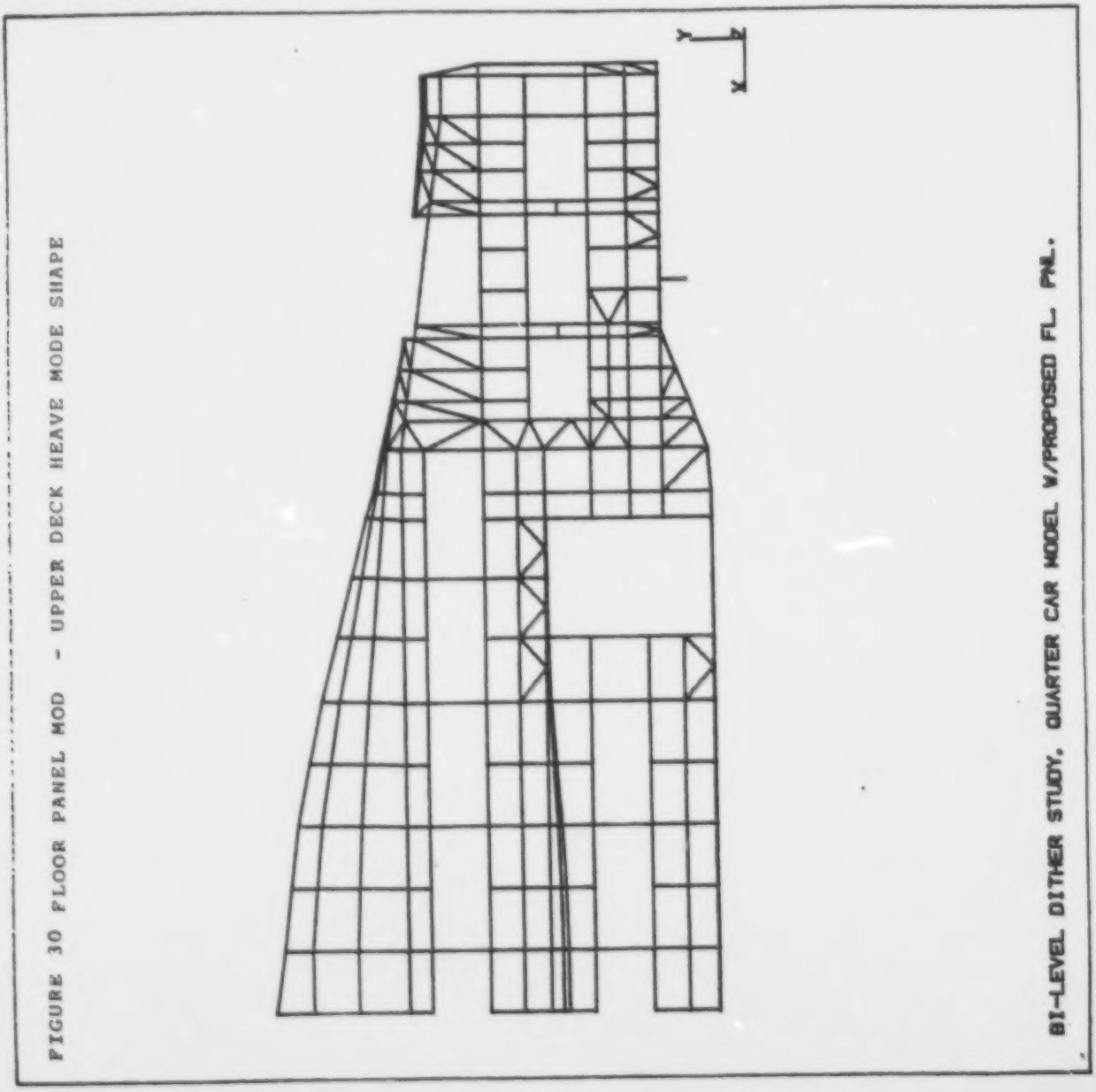


FIGURE 30 FLOOR PANEL MOD - UPPER DECK HEAVE MODE SHAPE

ANSYS 4.2
NOV 10 1987
22:23:33
DISPL.
STEP=1
ITER=2
FREQ=12
ZV=-1
DIST=280
XF=265
YF=105
ZF=28

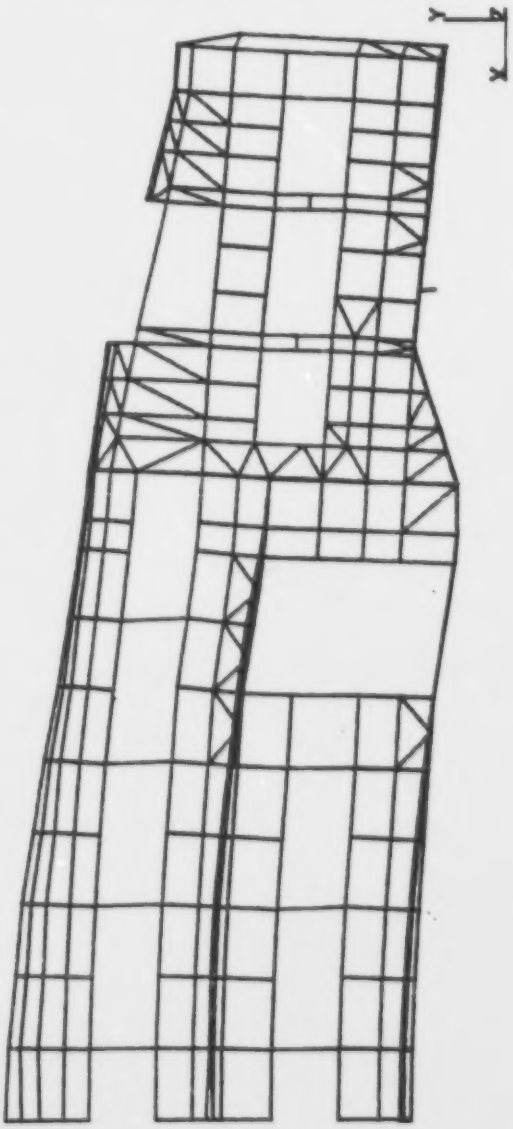


FIGURE 31 4 STANCHION MOD - FIRST BENDING MODE SHAPE

BILEVEL DITHER STUDY, QUARTER CAR MODEL W/STANCHION NO.5

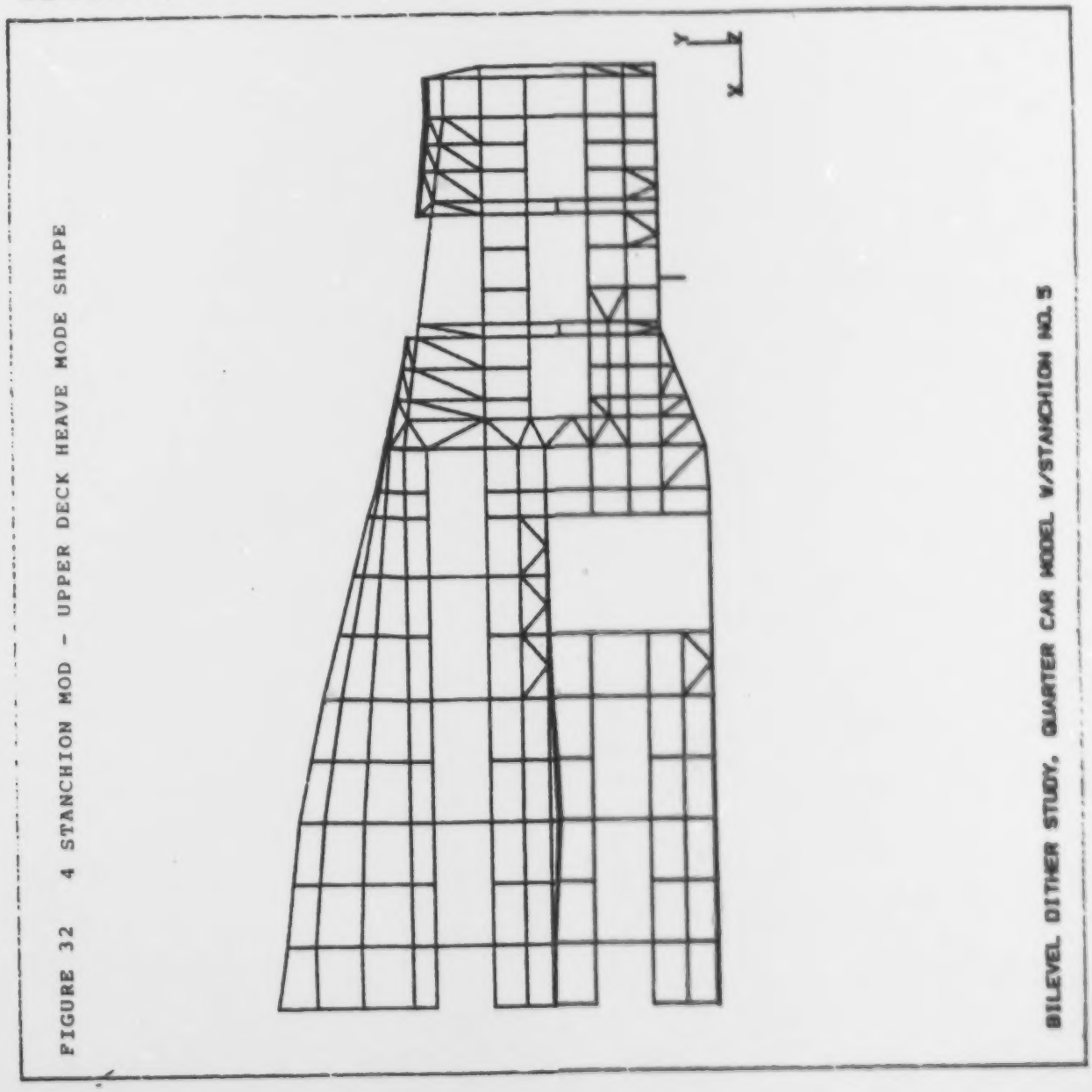


FIGURE 32 4 STANCHION MOD - UPPER DECK HEAVE MODE SHAPE

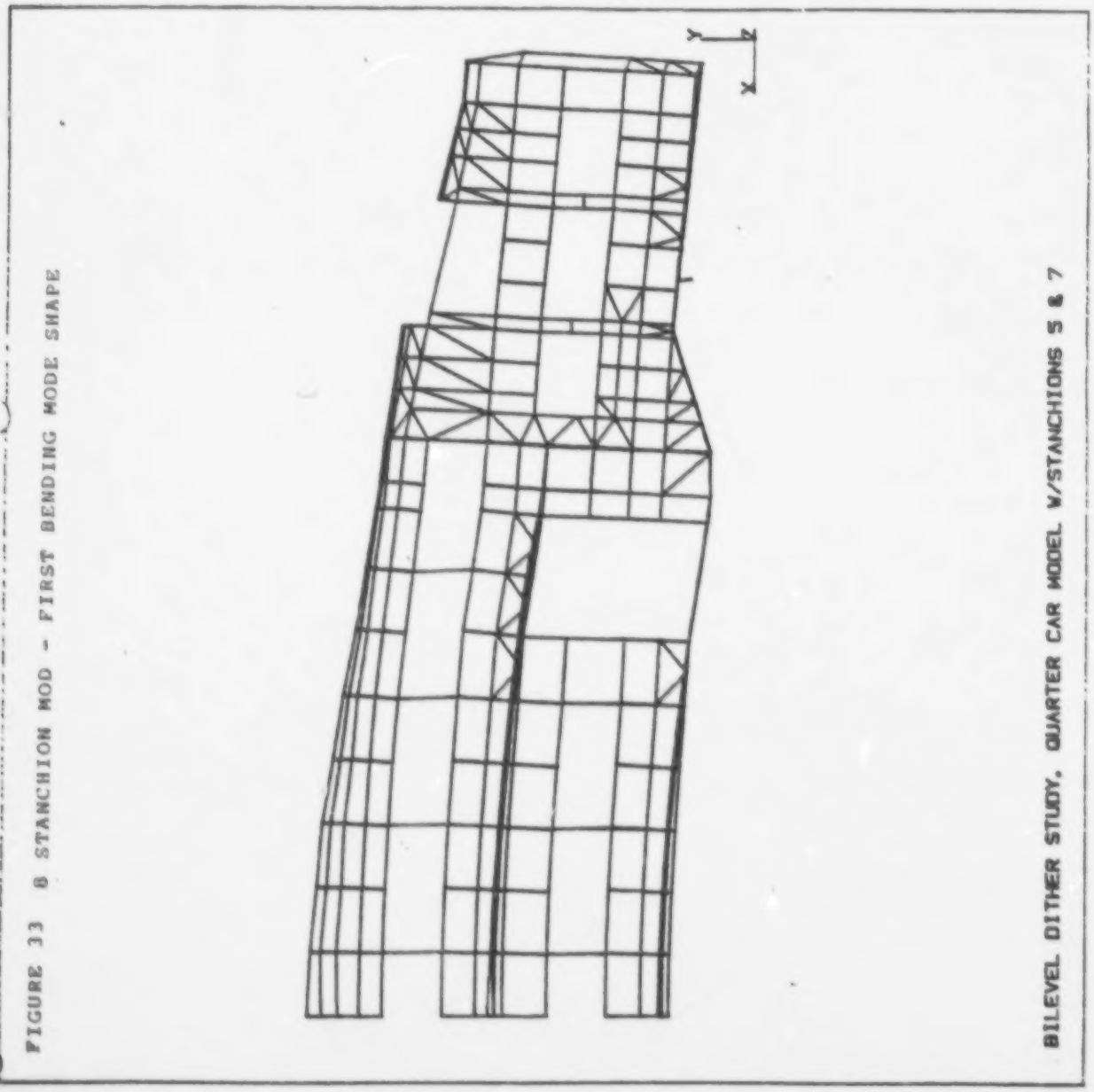


FIGURE 33 8 STANCHION MOD - FIRST BENDING MODE SHAPE

BILEVEL DITHER STUDY, QUARTER CAR MODEL W/STANCHIONS 5 & 7

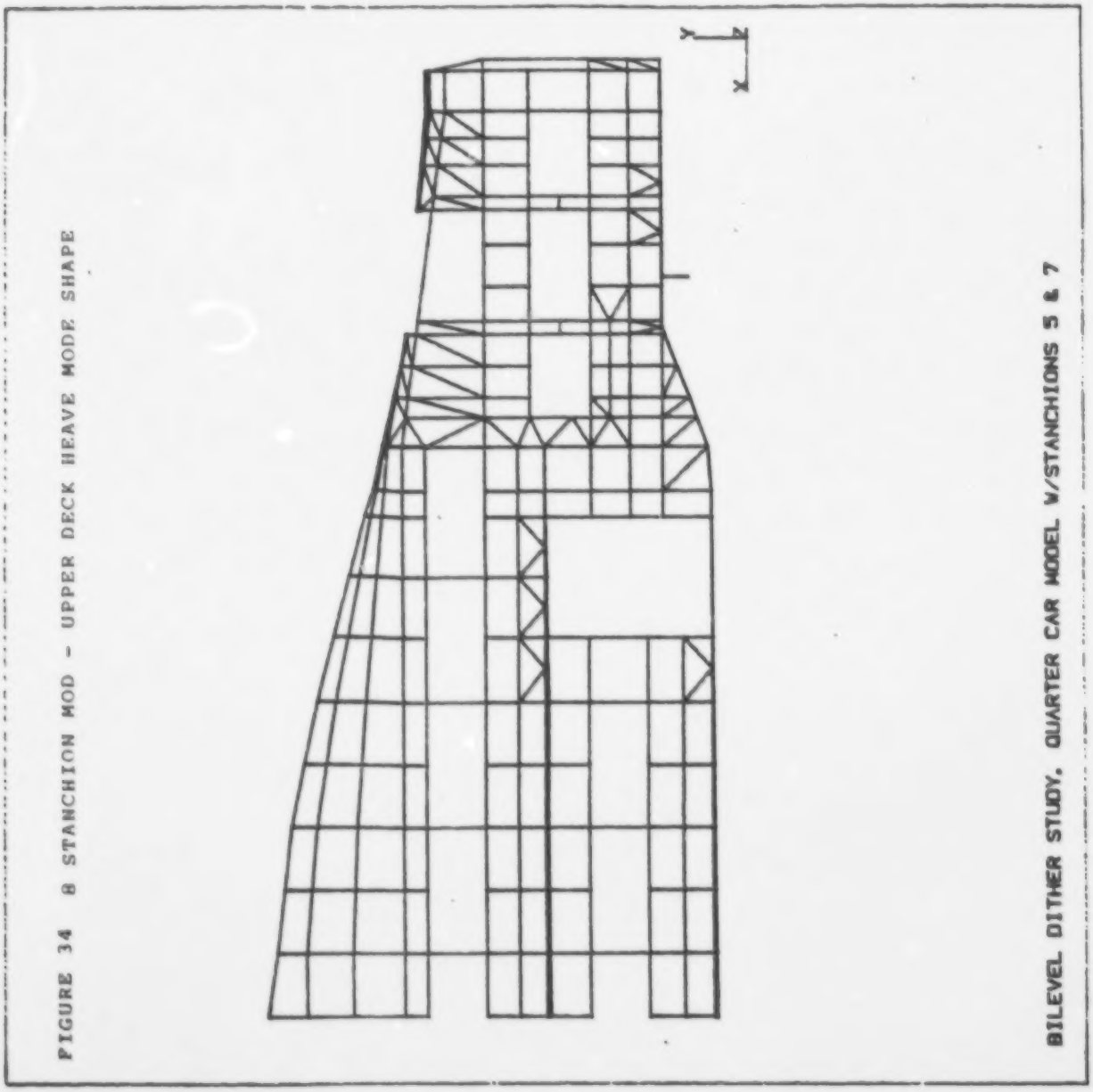


FIGURE 34 8 STANCHION MOD - UPPER DECK HEAVE MODE SHAPE

ANSYS 4.2
NOV 10 1987
23:58:15
DISPL.
STEP=1
ITER=2
FREQ=12.1
ZV=-1
DIST=280
XF=265
YF=105
ZF=28

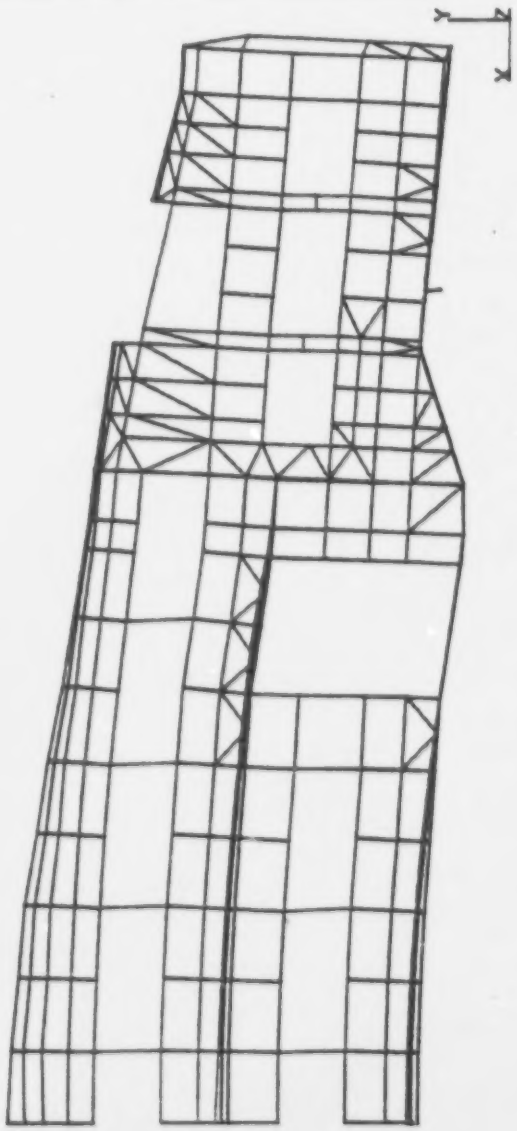


FIGURE 35 CROSS BEAM MOD - FIRST BENDING MODE SHAPE

QUARTER CAR DYNAMIC MODEL WITH BOXED-IN UPPER FLOOR BEAMS

ANSYS 4.2
NOV 20 1987
0:36: 5
DISPL
STEP=1
ITER=3
FREQ=15.8
ZV=-1
DIST=280
XF=265
YF=105
ZF=20

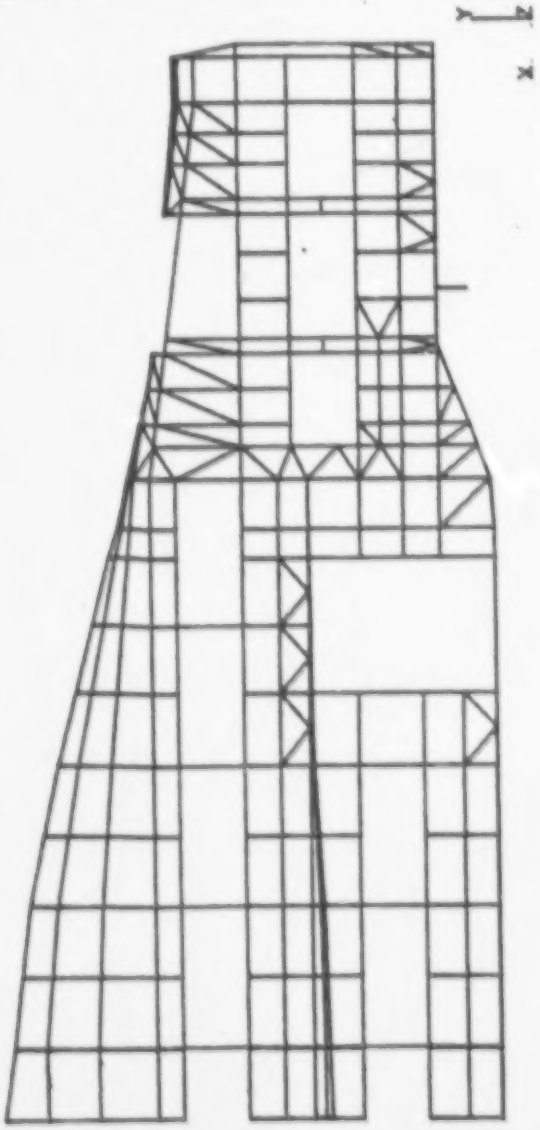


FIGURE 36 CROSS BEAM MOD - UPPER DECK HEAVE MODE SHAPE

QUARTER CAR DYNAMIC MODEL WITH BOXED-IN UPPER FLOOR BEAMS

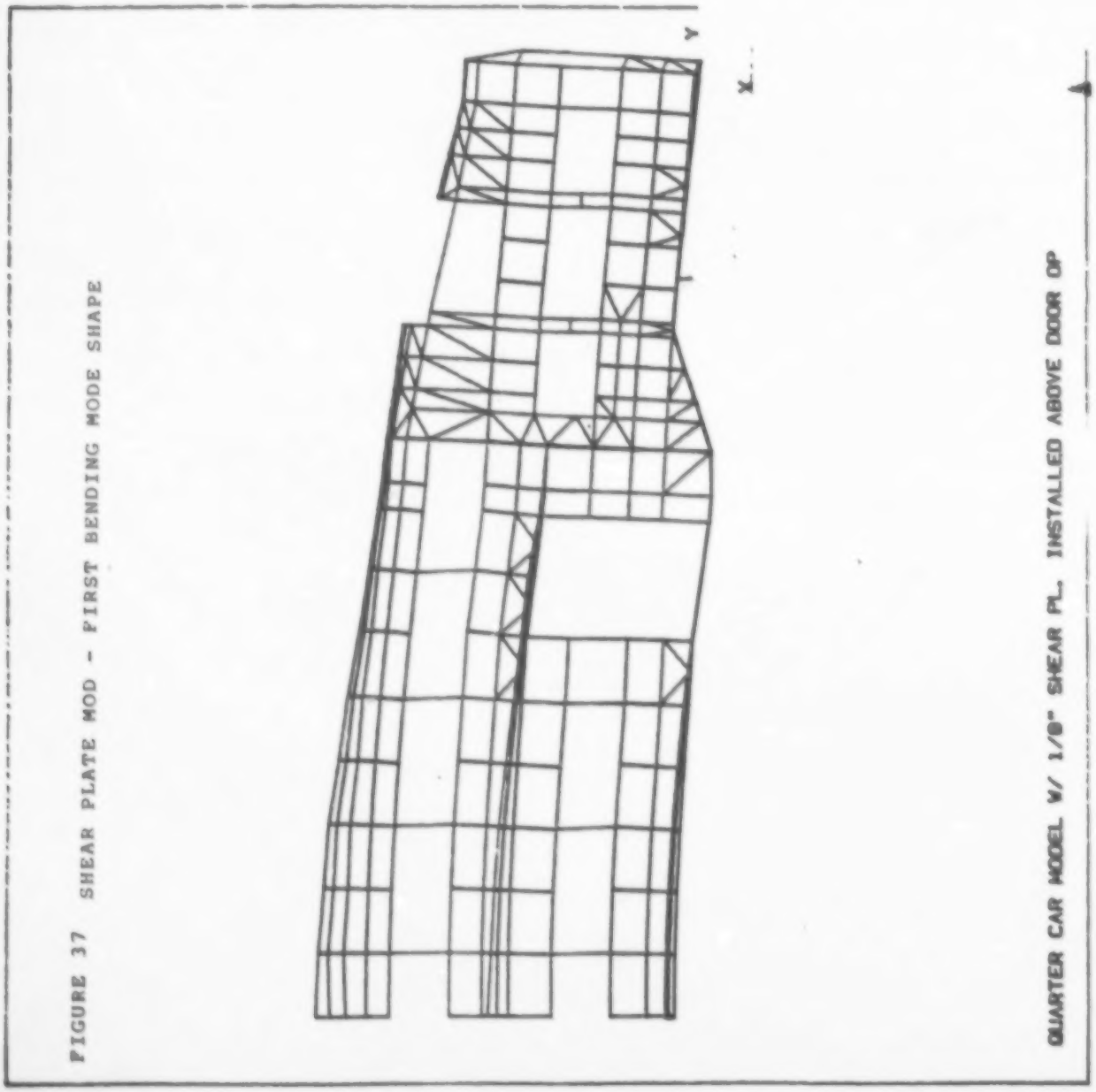


FIGURE 37 SHEAR PLATE MOD - FIRST BENDING MODE SHAPE

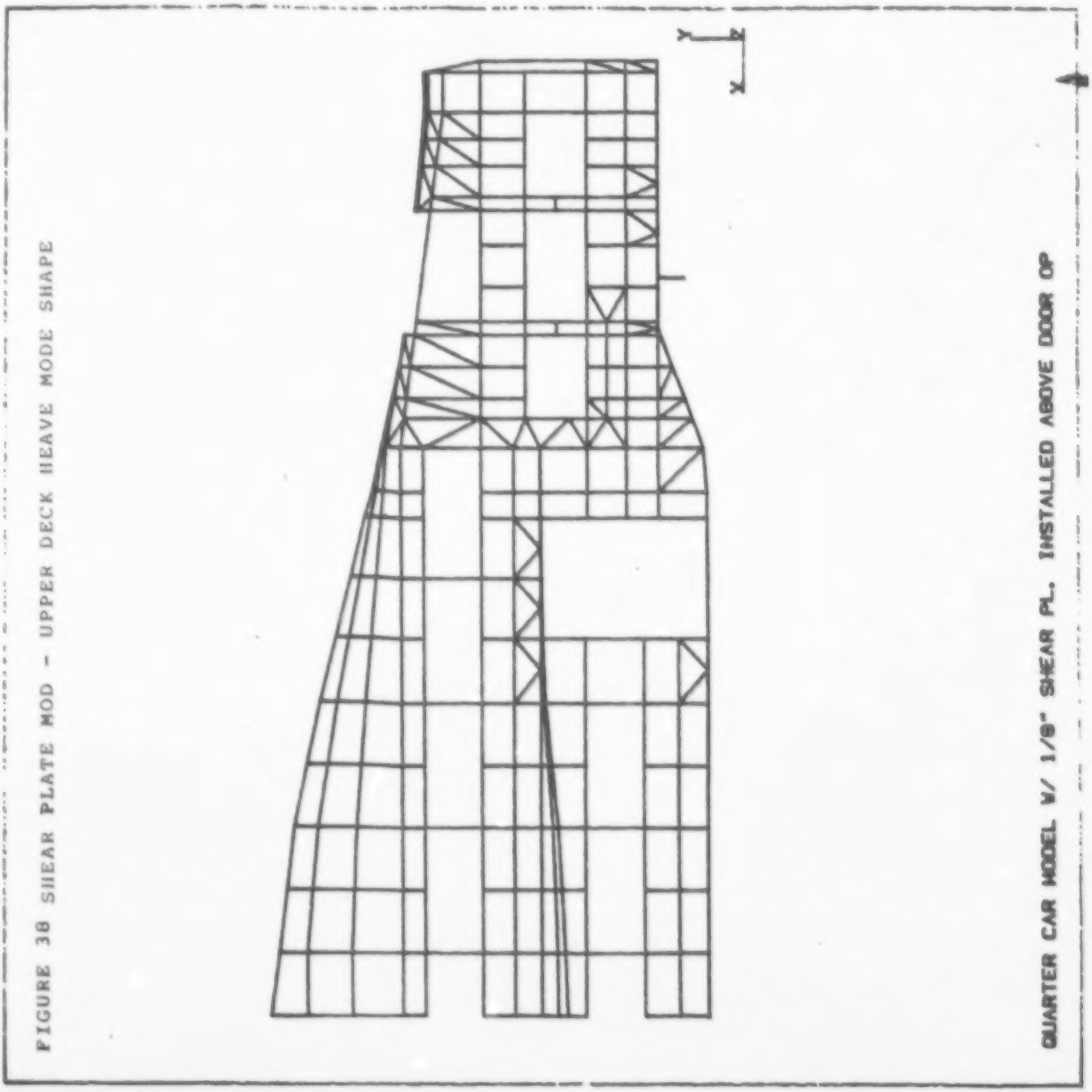


FIGURE 38 SHEAR PLATE MOD - UPPER DECK HEAVE MODE SHAPE

APPENDIX A

TRUCK SUSPENSION PARAMETERS

(SUPPLIED BY DOFASCO [8])

BI-LEVEL II TRUCK SUSPENSION PARAMETERS

1. Dimensions (Inches)

Lateral Distance Between:	- primary springs	46.4
	- sidebearers	44.0
	- snubbers	67.0
	- air springs	88.0
	- secondary dampers	104.0
Longitudinal Distance Between:	- truck centres	384.0
	- wheelsets	102.0
	- vertical damper and truck centre	13.0
	- sidebearers	20.0
	- lateral damper and truck centre	14.5
Height of:	- carbody c.g. (empty)	81.0
	- bolster c.g.	22.3
	- truck frame c.g.	17.1
	- wheelset c.g.	16.5
	- chevrons	16.5
	- lateral damper	27.4
	- sidebearers	18.9
	- traction pads	16.8
	- snubbers	12.0
	- airspring (AW0)	33.5

2. Weights (lbs.)

Carbody - empty	80,720 (no passengers)
- loaded	105,830 (162 passengers)
- crush	123,345 (275 passengers)
- super crush	148,610 (438 passengers)
Bolster	2,200
Frame	5,750
Wheelset	2,750

3. Mass Moments of Inertia (lb. in. sec.^{2})

Carbody	- pitch	20.0×10^6
	- roll	0.7×10^6
	- yaw	20.0×10^6
Bolster	- pitch	300
	- roll	6,900
	- yaw	5,400
Frame	- pitch	15,700
	- roll	9,400
	- yaw	27,100
Wheelset	- pitch	500
	- roll	5,100
	- yaw	5,100

4. Spring Stiffnesses (lb. in.)

Chevrons (per journal)	- vertical	11,300
	- lateral	40,400
	- longitudinal	143,000
Sidebearer (each)	- vertical	40,000
	- lateral	720
	- longitudinal	290
Traction Pads (per pair)	- vertical	nil
	- lateral	300
	- longitudinal	66,000
Air spring (each)	- vertical	5,500

Note: All spring rates are static values based on the empty car weight.

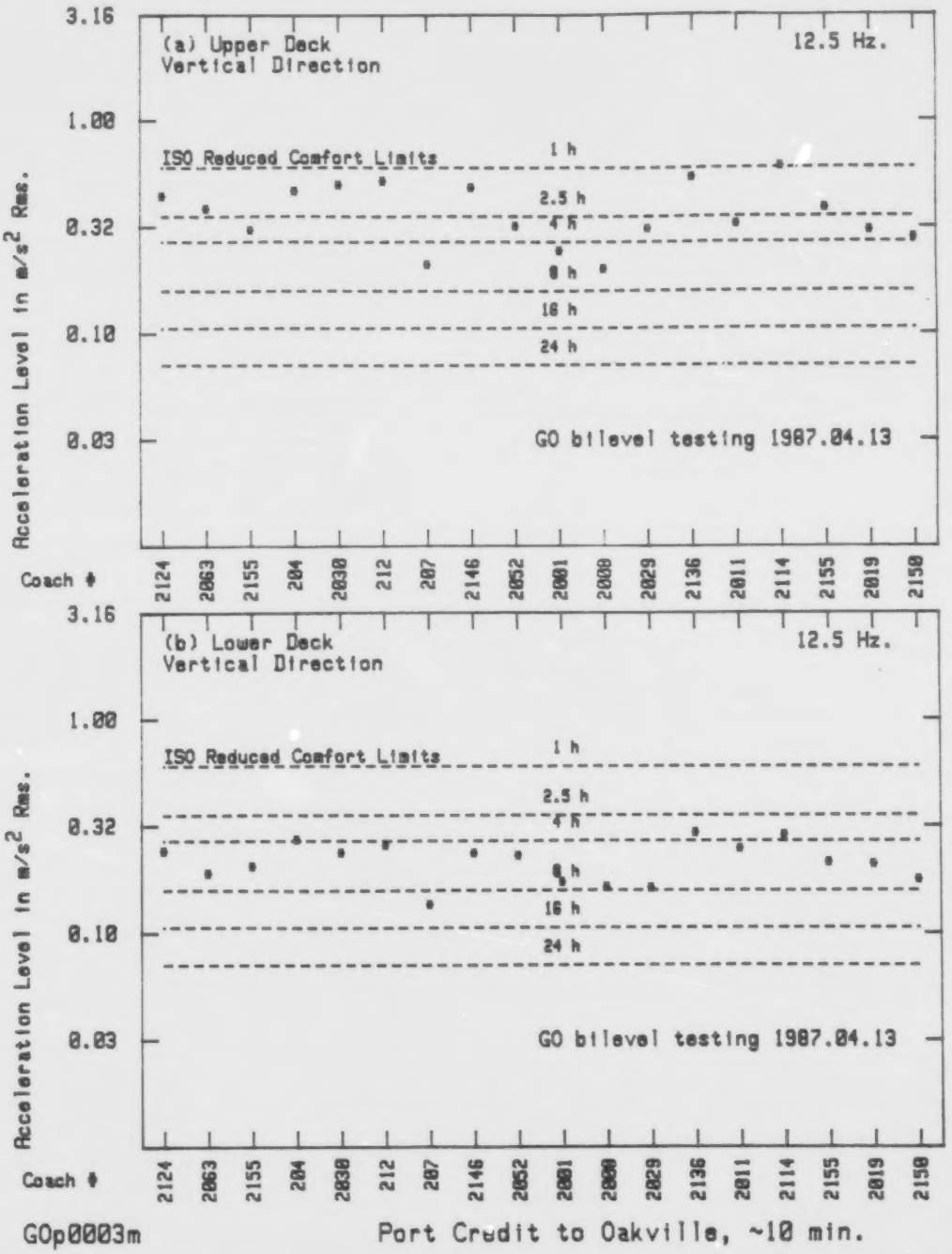
5. Damping Rates

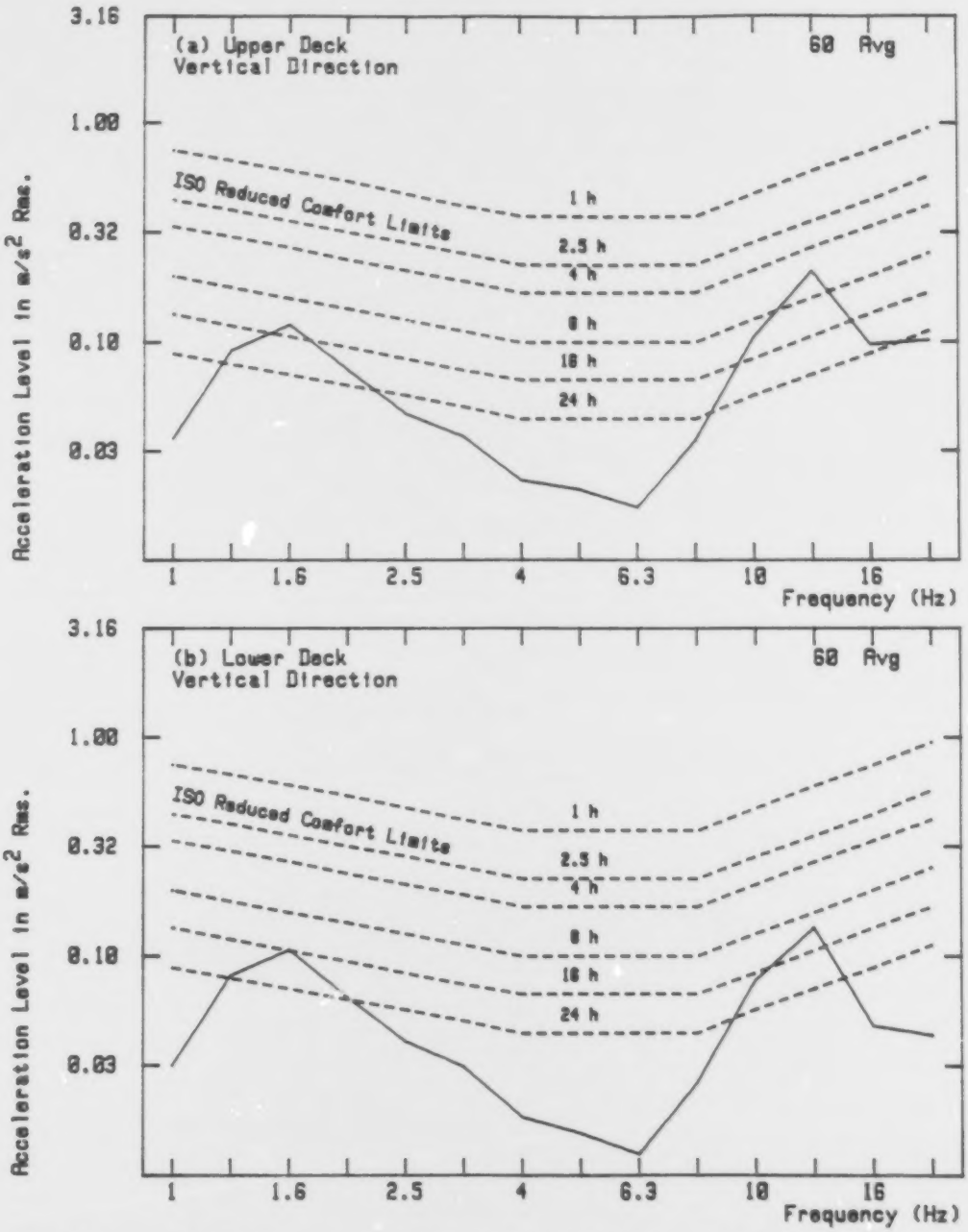
Secondary vertical (each)	189 lb. sec./in.
Secondary lateral (each)	252 lb. sec./in.
Friction Snubber (each)	2,050 lb.

APPENDIX B

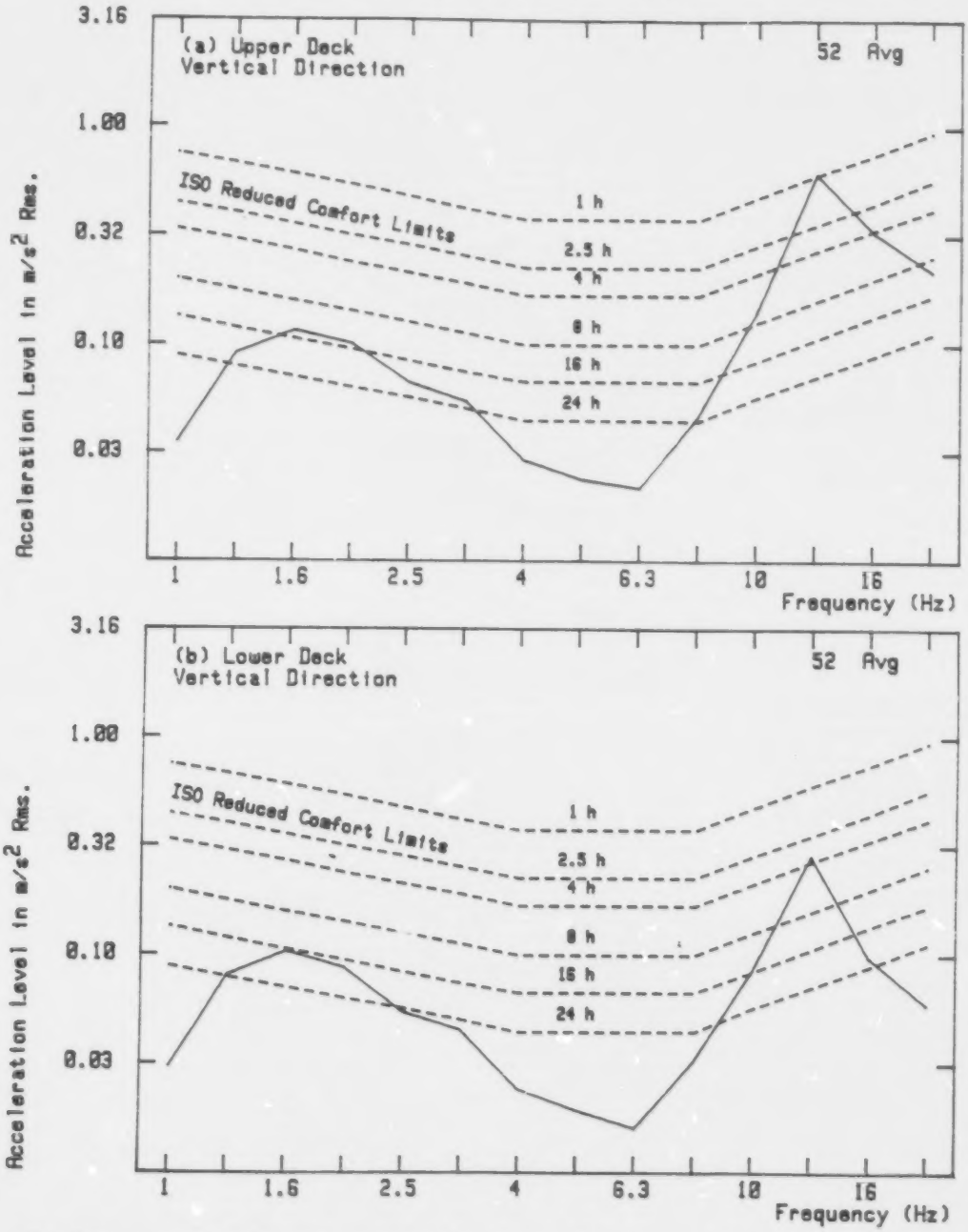
BASELINE BILEVEL RIDE DATA

SOURCE: MTO TESTS 1987 APRIL

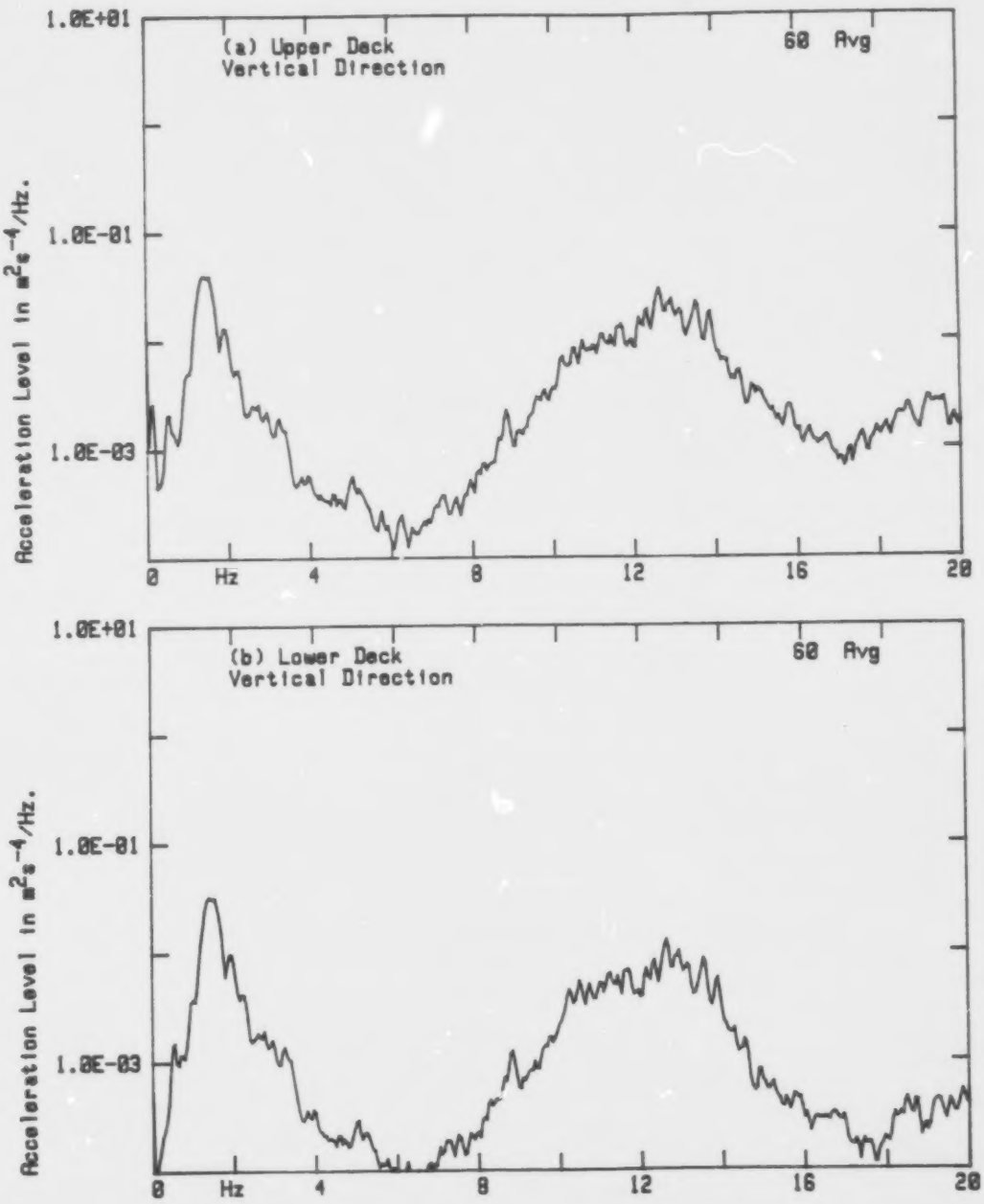

(a) Upper Deck
Vertical Direction
12.5 Hz.
ISO Reduced Comfort Limits
1 h
2.5 h
4 h
8 h
16 h
24 h
GO bilevel testing 1987.04.13
Acceleration Level in m/s² Rms.
3.16
1.00
0.32
0.10
0.03
Coach #
2124 2063 2155 204 2030 212 207 2146 2052 2001 2000 2029 2136 2011 2114 2155 2019 2150
(b) Lower Deck
Vertical Direction
12.5 Hz.
ISO Reduced Comfort Limits
1 h
2.5 h
4 h
8 h
16 h
24 h
GO bilevel testing 1987.04.13
Acceleration Level in m/s² Rms.
3.16
1.00
0.32
0.10
0.03
Coach #
2124 2063 2155 204 2030 212 207 2146 2052 2001 2000 2029 2136 2011 2114 2155 2019 2150
GOp0003m
Port Credit to Oakville, ~10 min.



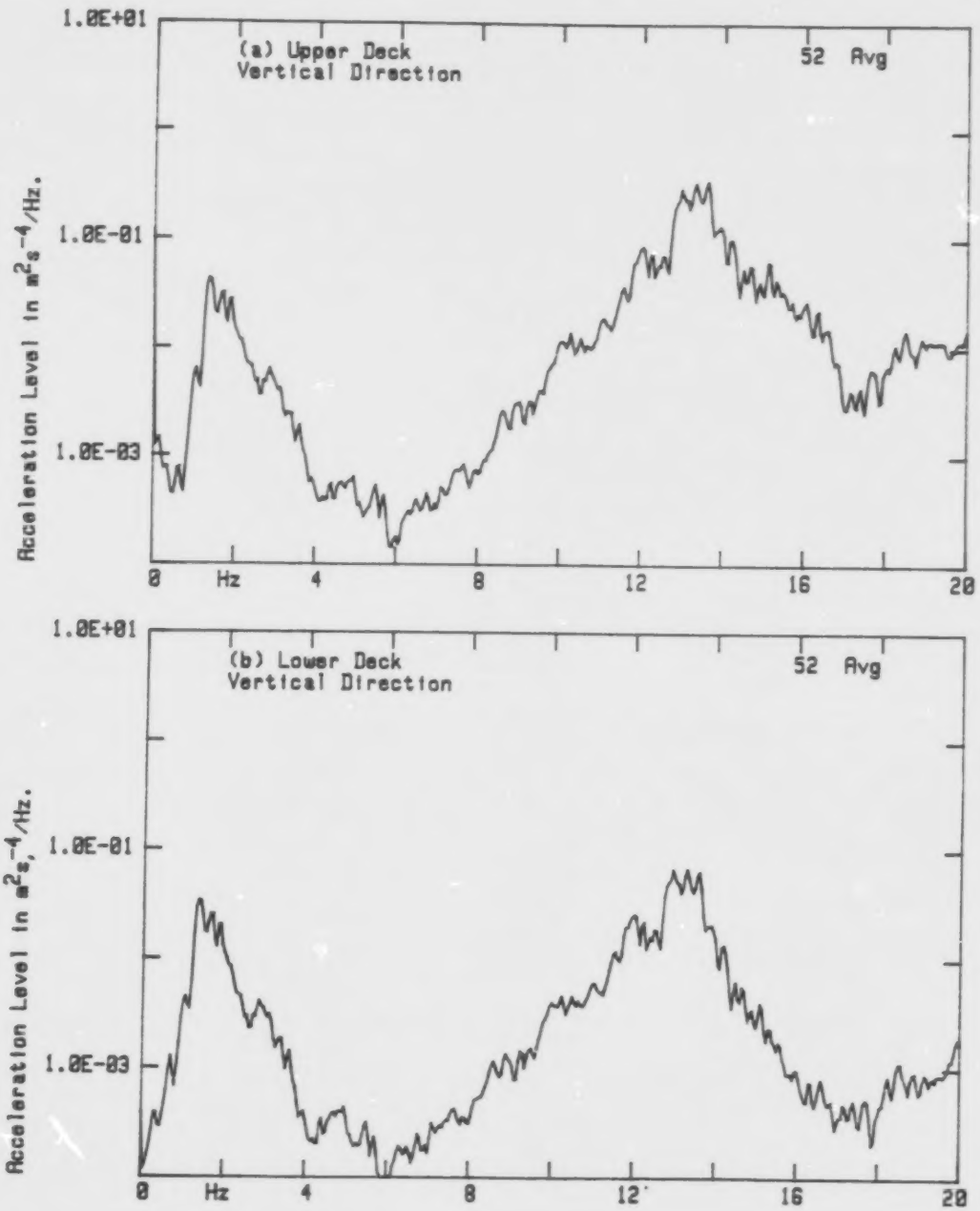
G0p0002m, G007d02 Coach 207, Port Credit to Oakville,~10 min
GO bilevel testing 1987.04.09



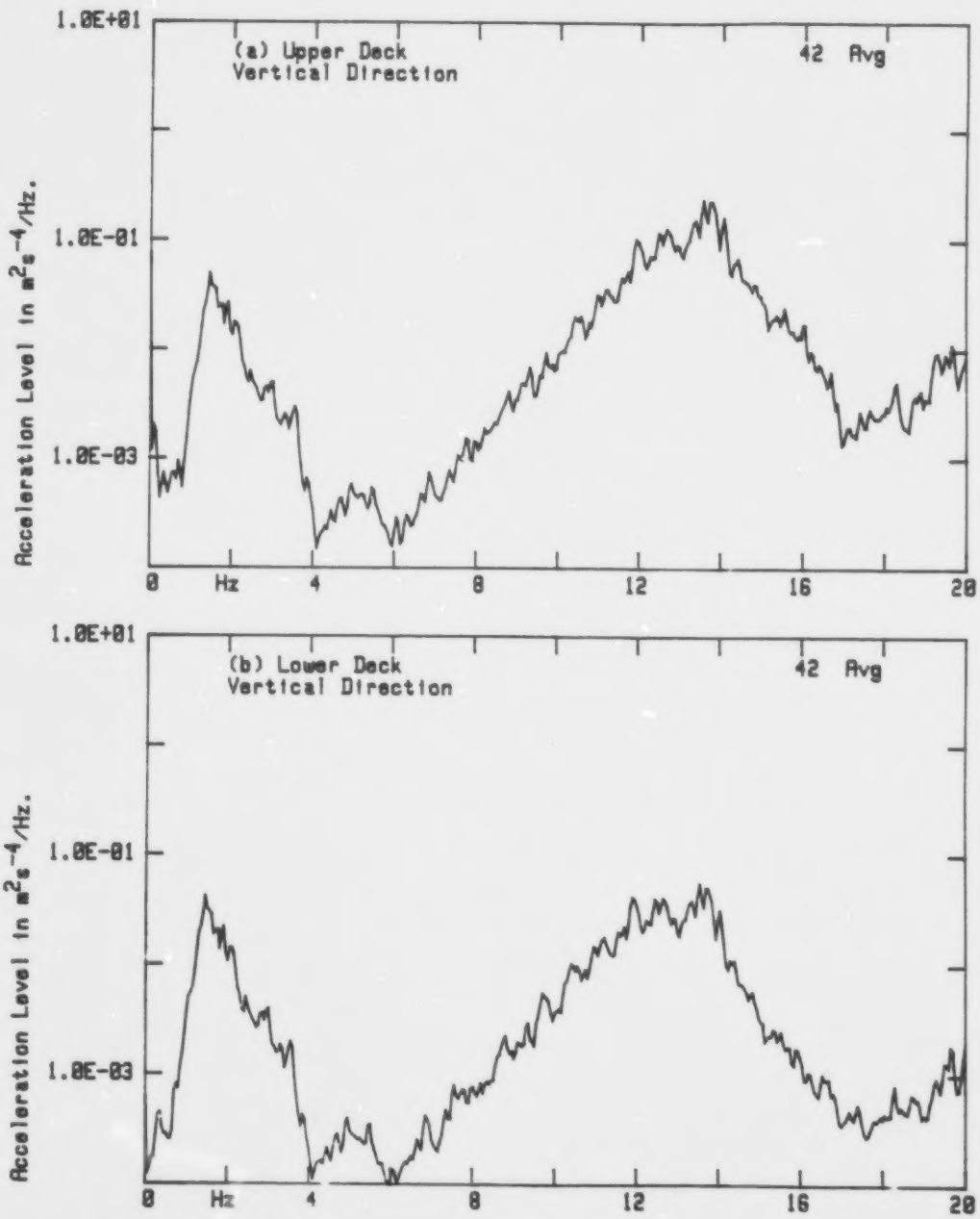
GOp0002m, GO15d02 Coach 2114, Port Credit to Oakville,~10 min
GO bilevel testing 1987.04.13



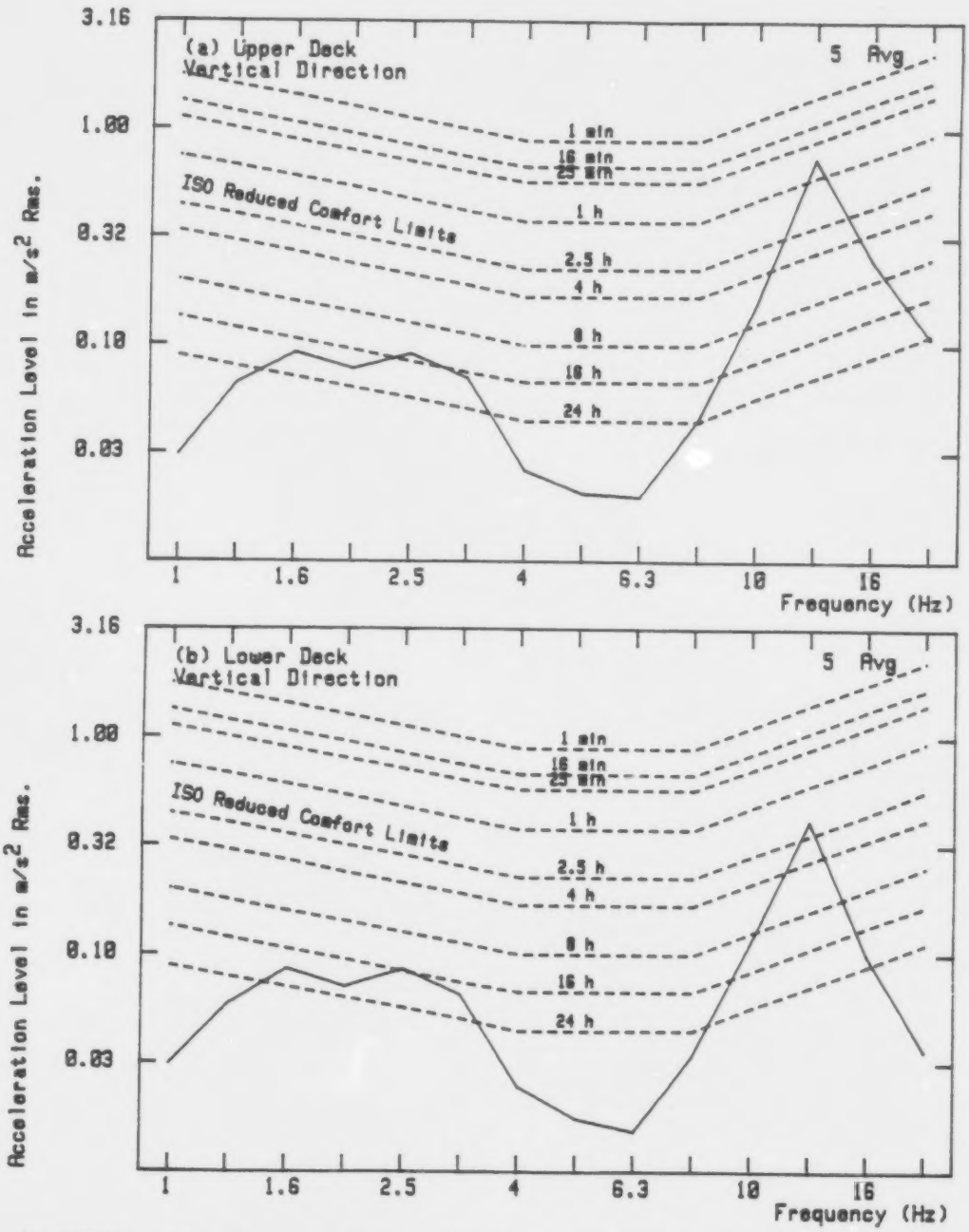
GOp0001m, G007d02 Coach 207, Port Credit to Oakville,~10 min
GO bilevel testing 1987.04.09



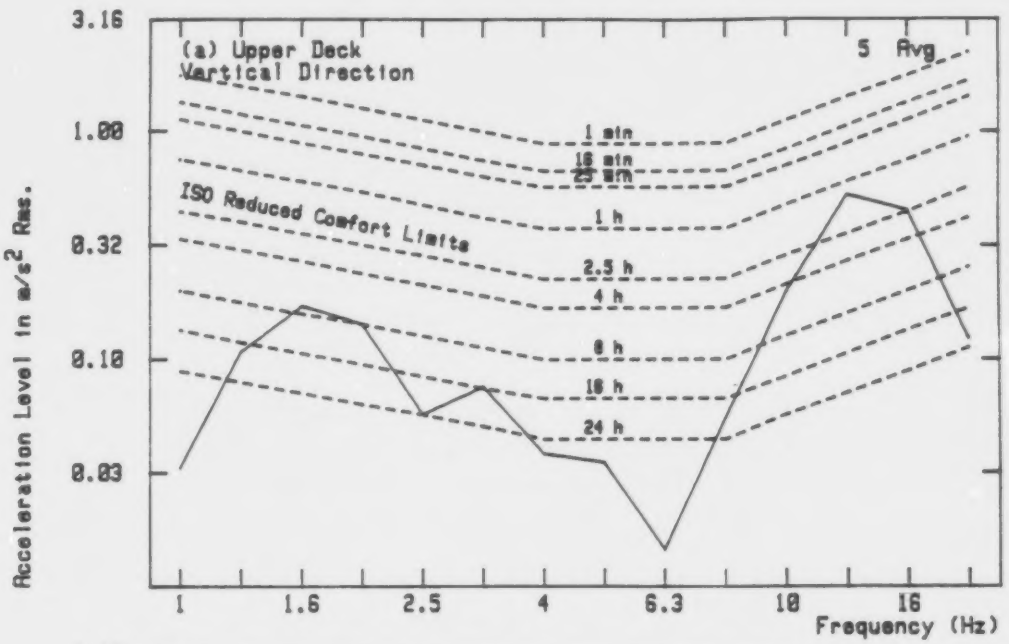
GOp0001m, GO15d02 Coach 2114, Port Credit to Oakville,~10 min
GO bilevel testing 1987.04.13

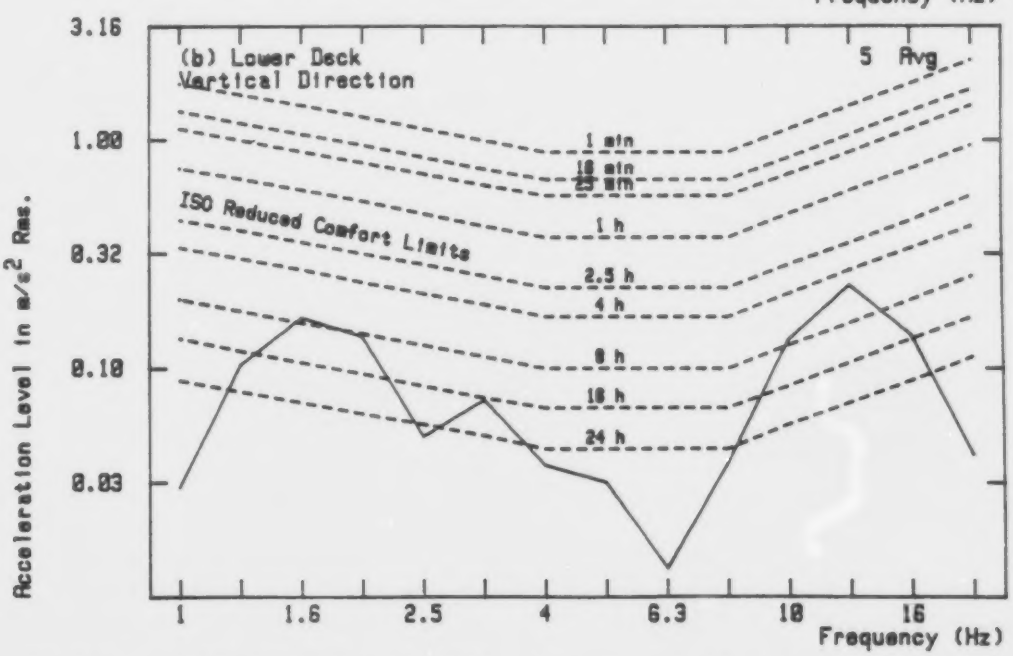


GOp0001m, G013d02 Coach 2136, Port Credit to Oakville,~10 min
GO bilevel testing 1987.04.18

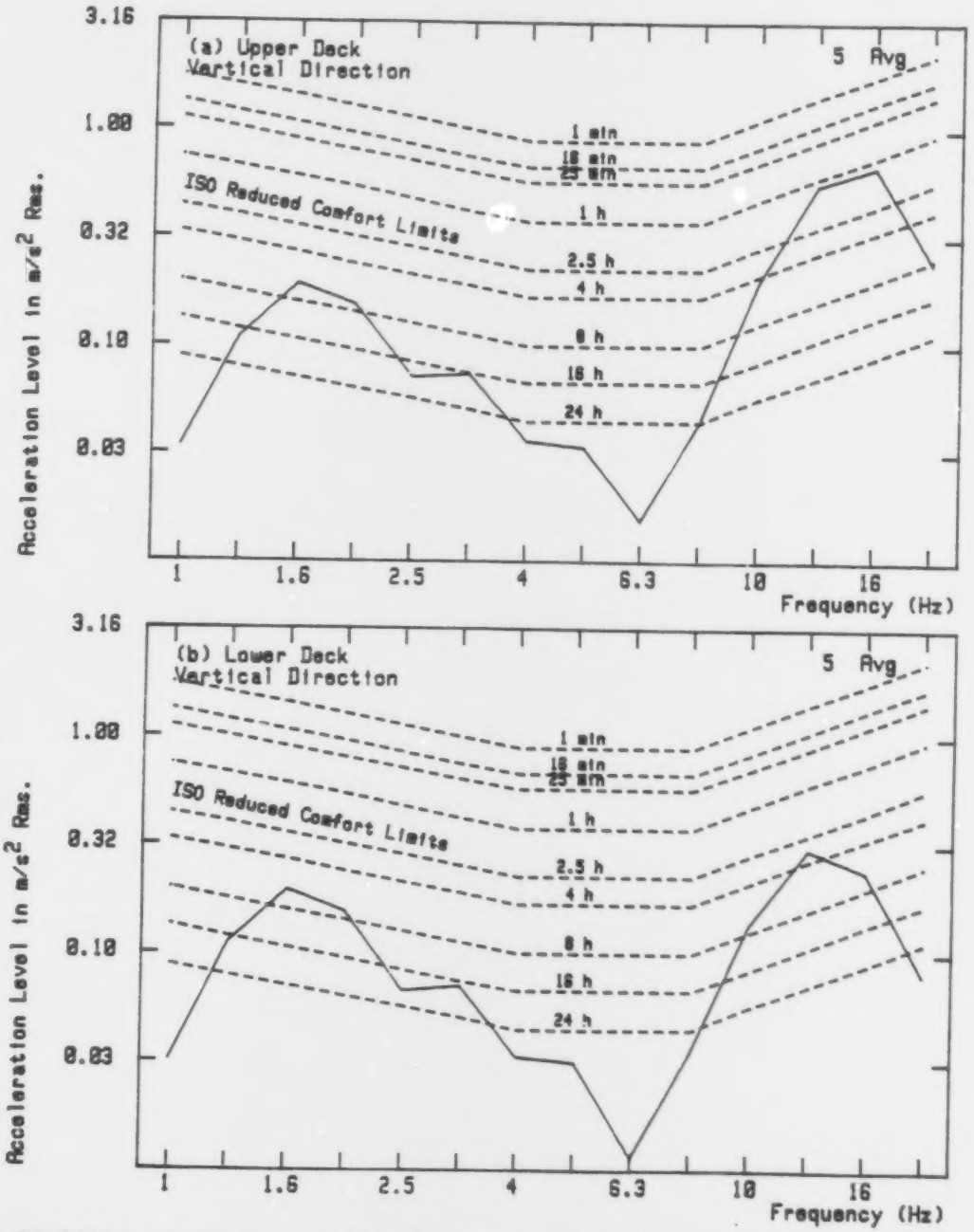


G0p0006m, G001e02 Coach 2124, Port Credit to Clarkson, 1min
G0 bilevel testing 1987.04.08

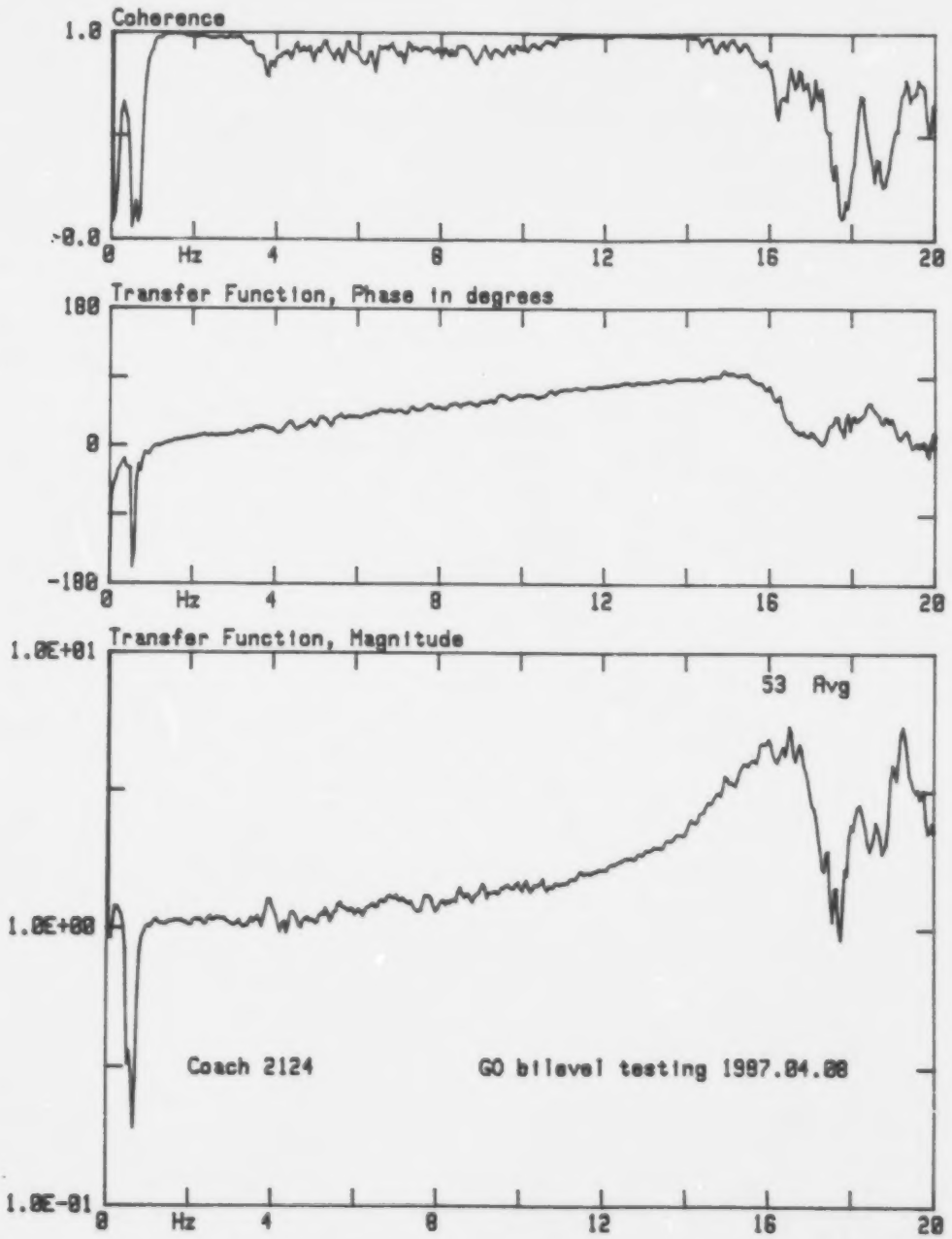




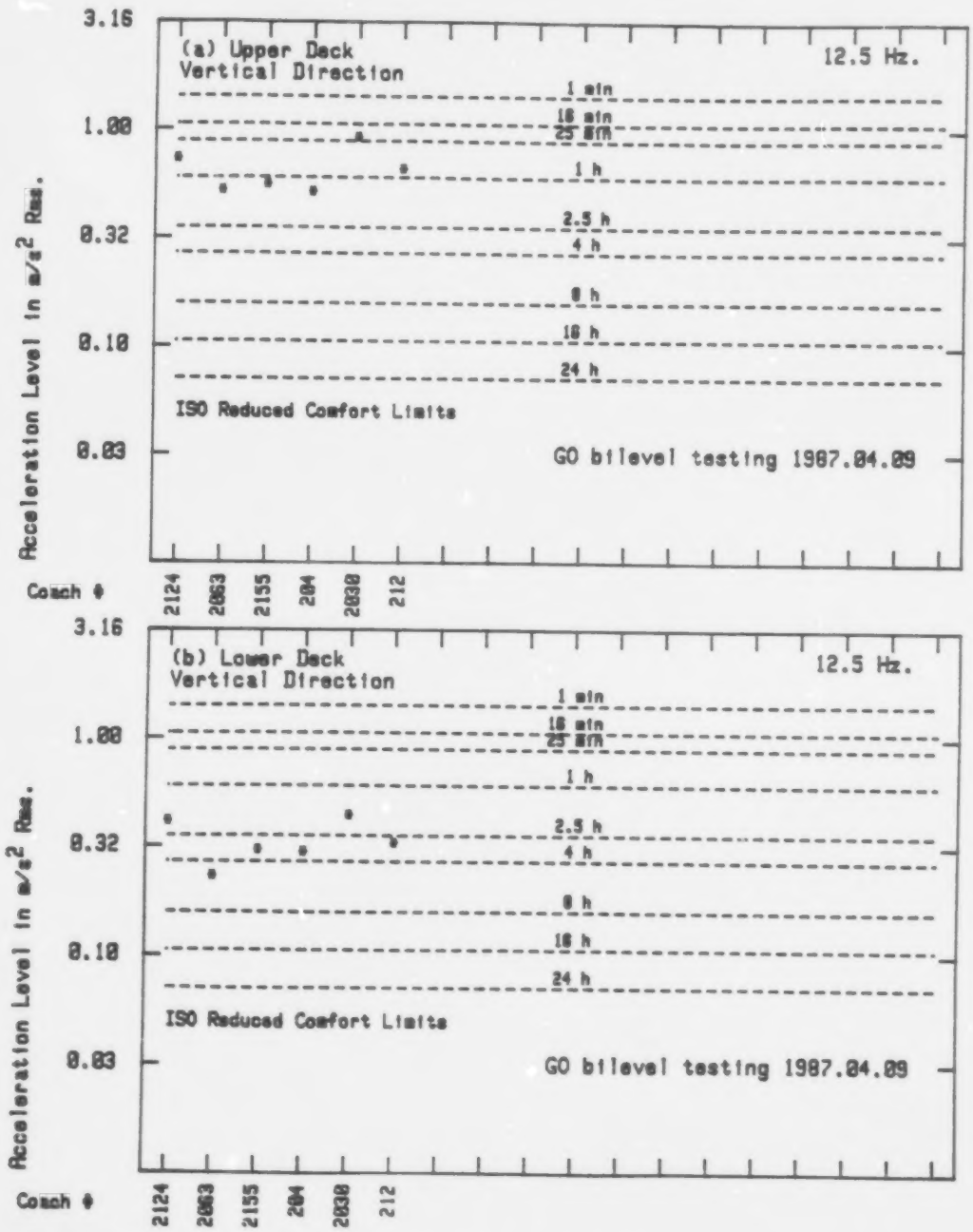
GOp0006m, GO02e02 Coach 2063, Clarkson to Oakville, 1min
GO bilevel testing 1987.04.08



GOp0006m, G003e02 Coach 2155, Clarkson to Oakville, 1min
GO bilevel testing 1987.04.08

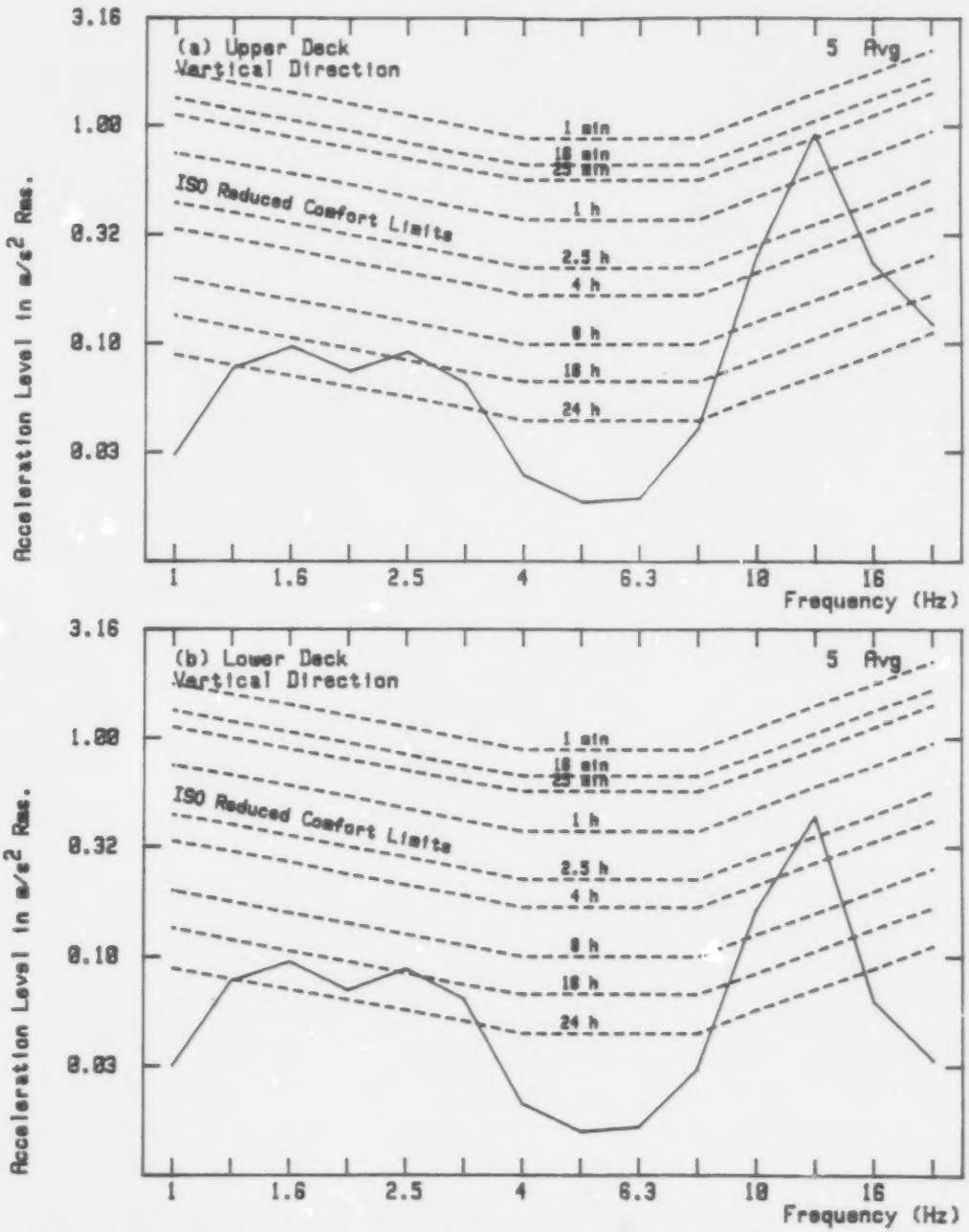


GOp0008m , G001d02, Transfer Function between Upper & Lower Deck

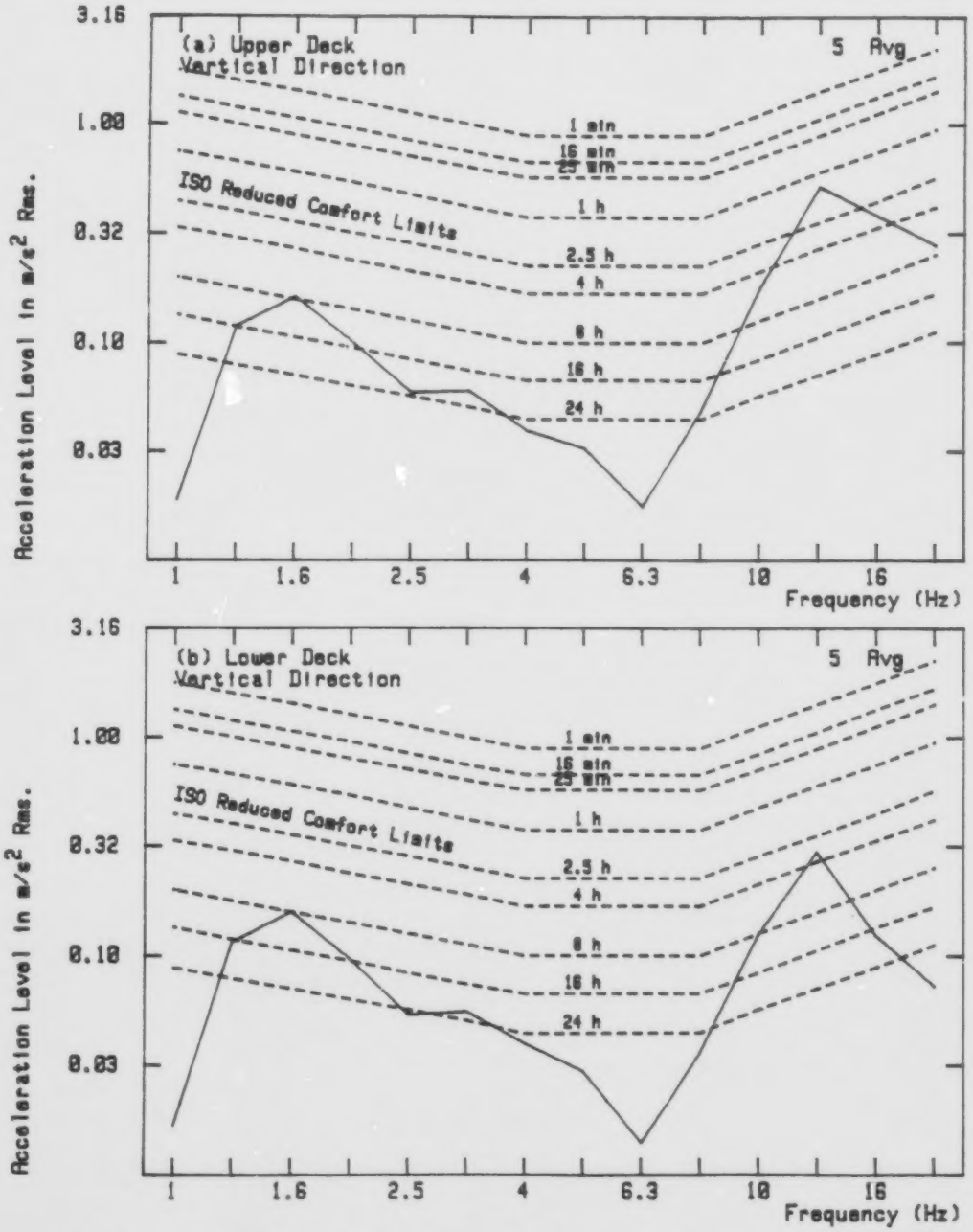


GOp0007m

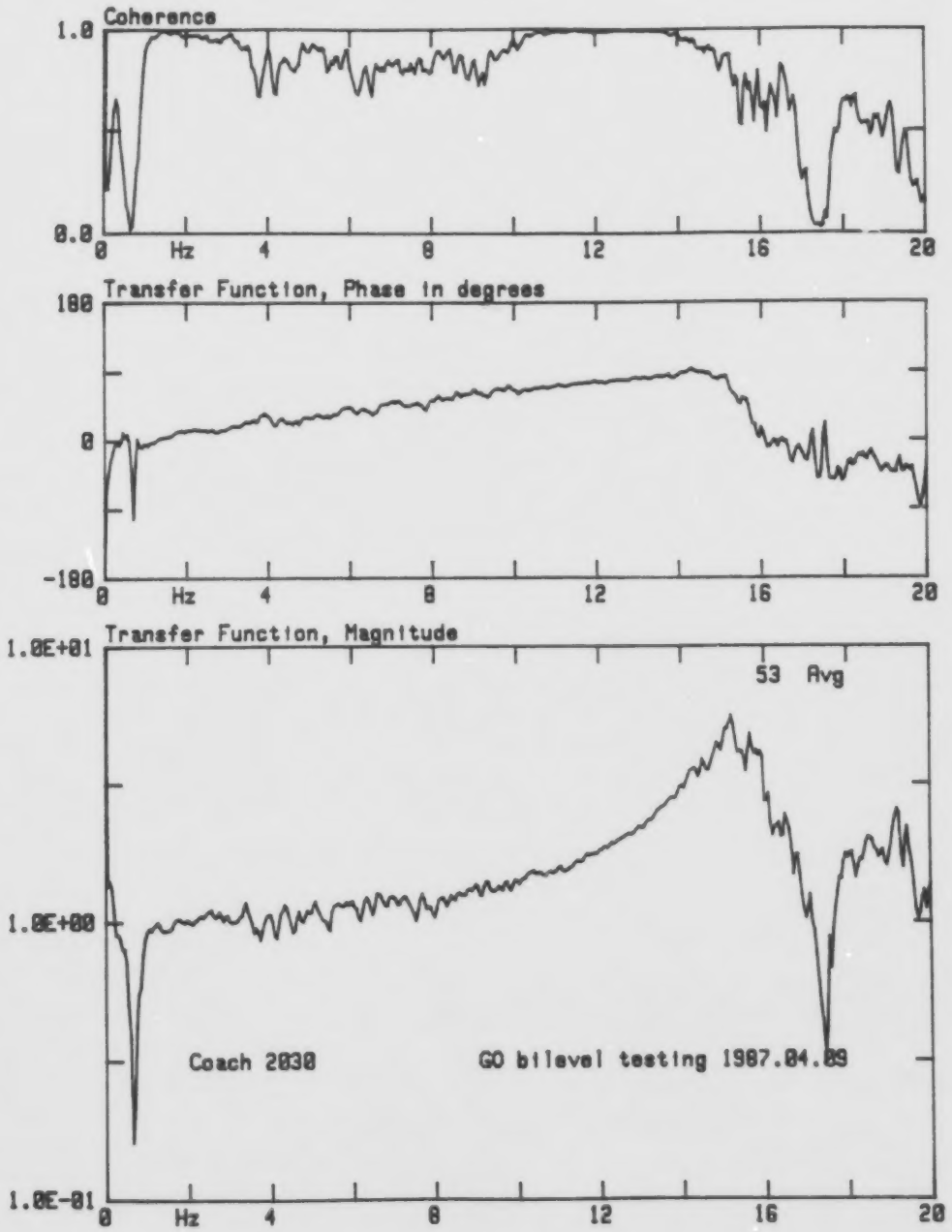
Worst Case for 1 min



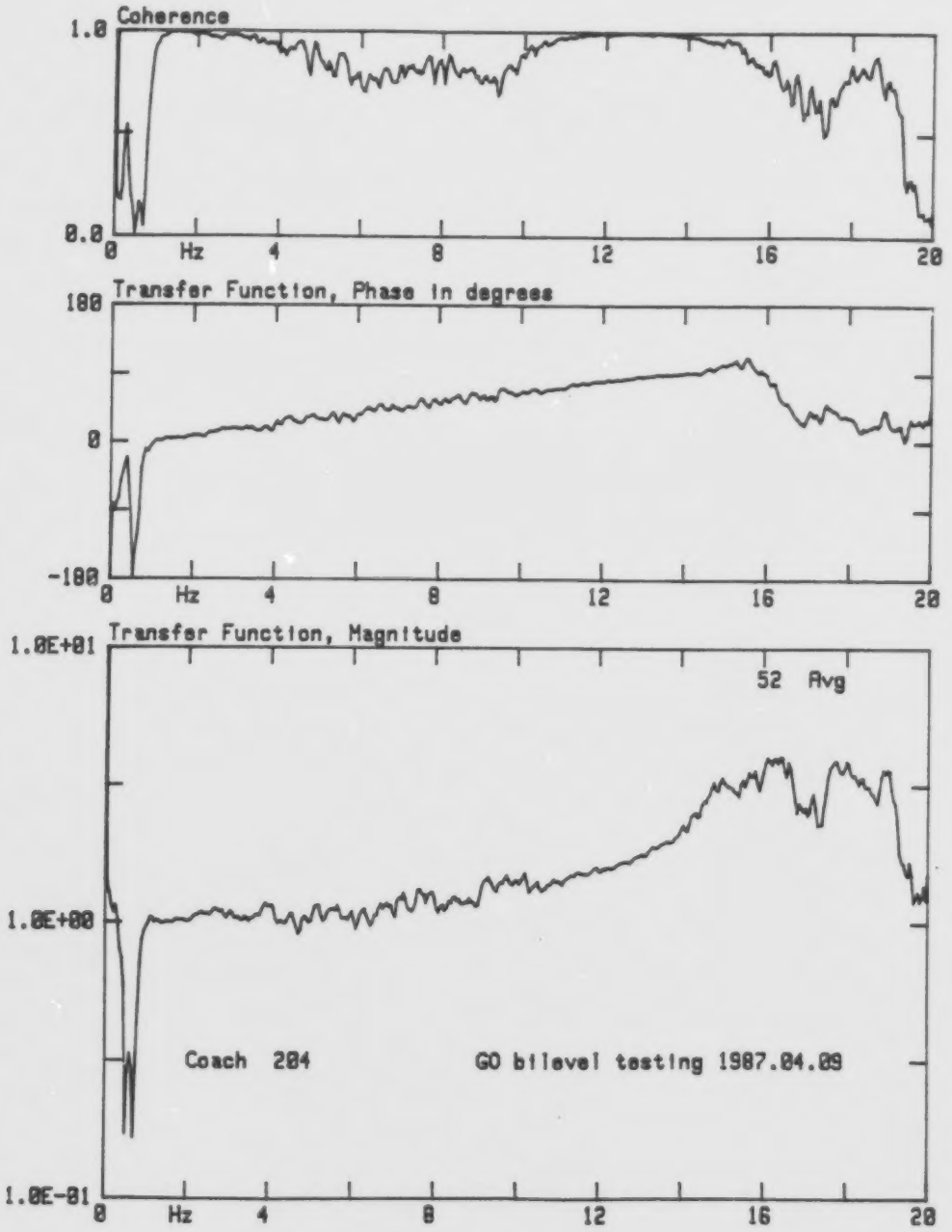
G0p0006m, G005e02 Coach 2030, Port Credit to Clarkson, 1min
GO bilevel testing 1987.04.09



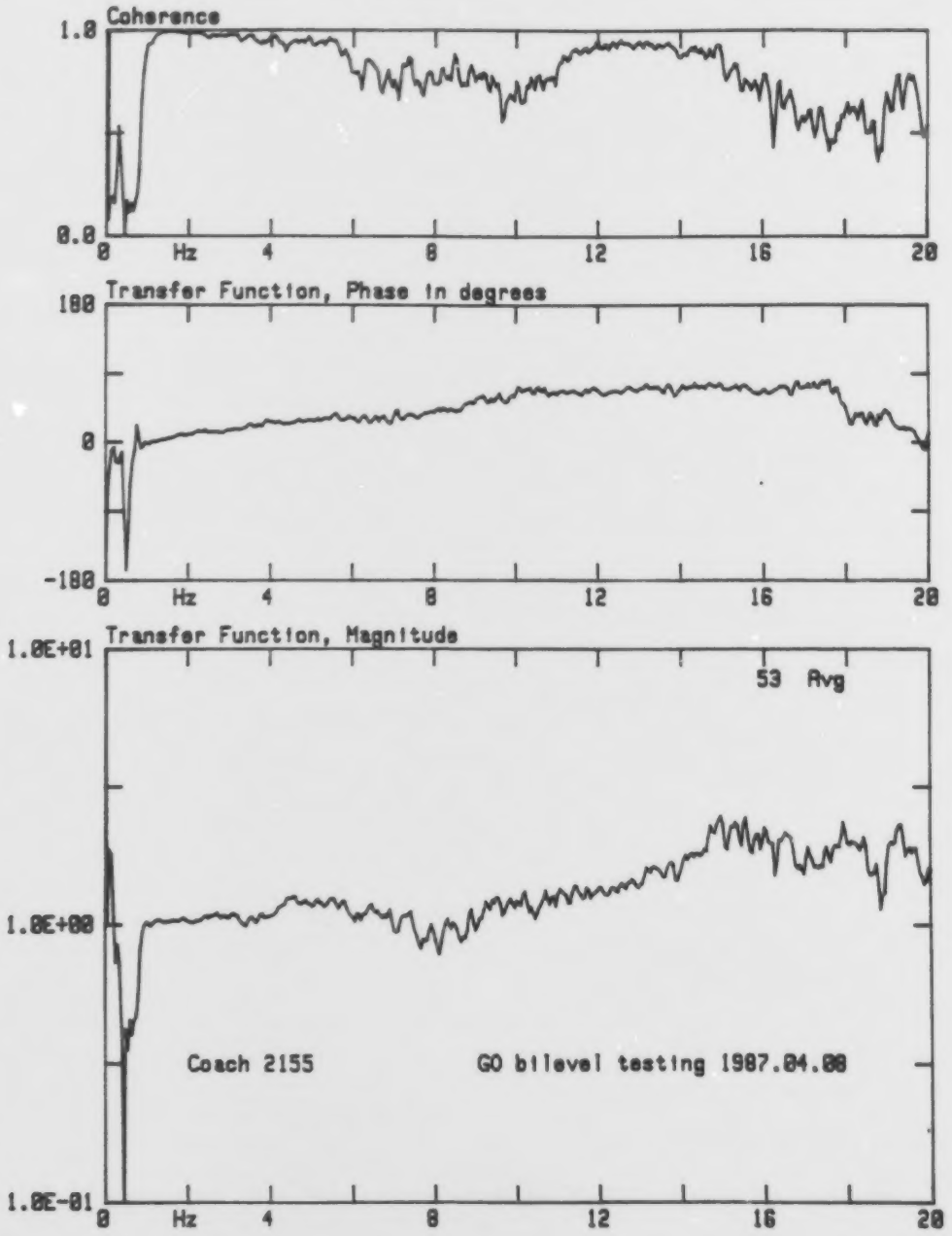
GOp0006m, GO04e02 Coach 204, Clarkson to Oakville, 1min
GO bilevel testing 1987.04.09



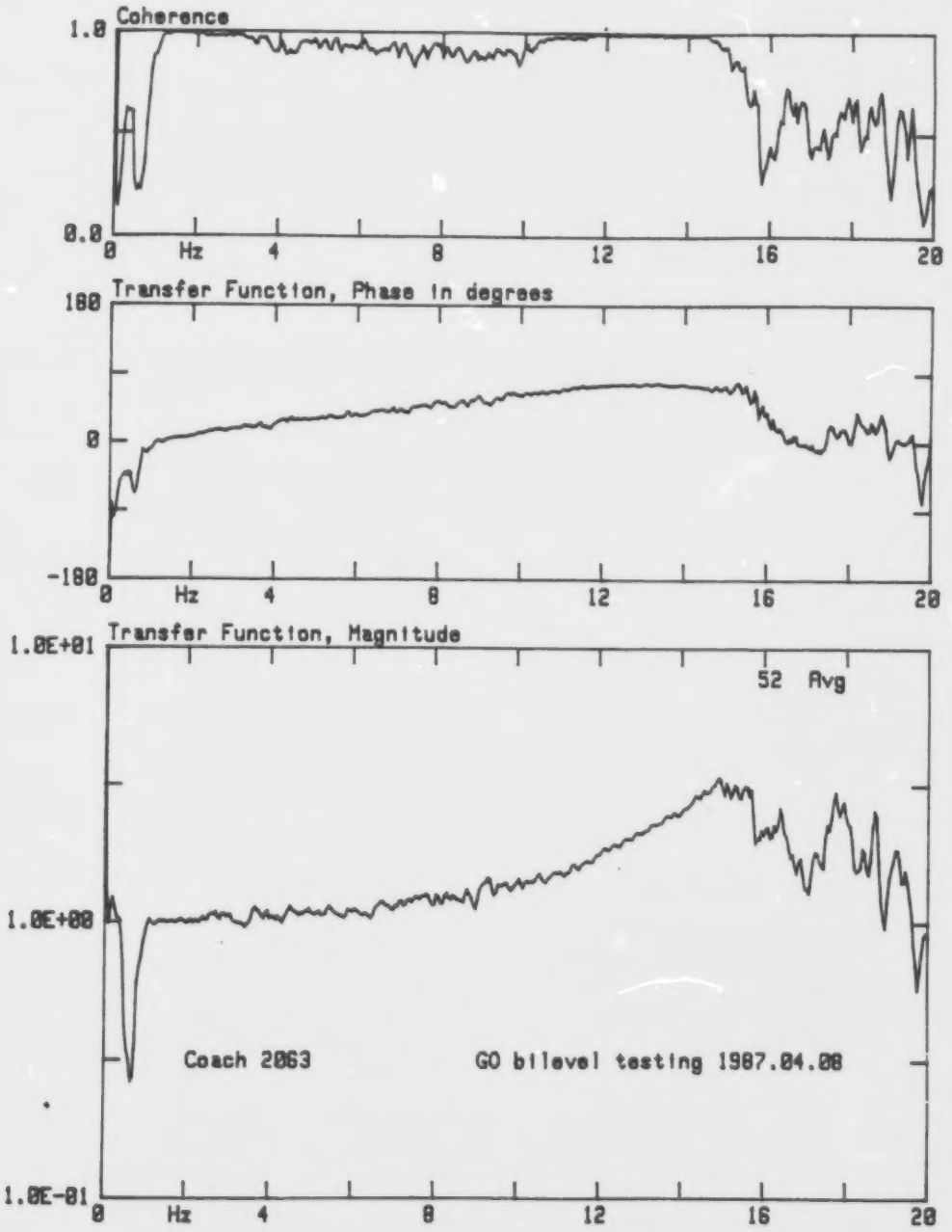
GOp0008m , G005d02, Transfer Function between Upper & Lower Deck



GOp0008m , G004d02, Transfer Function between Upper & Lower Deck



GOp0008m , G003d02, Transfer Function between Upper & Lower Deck



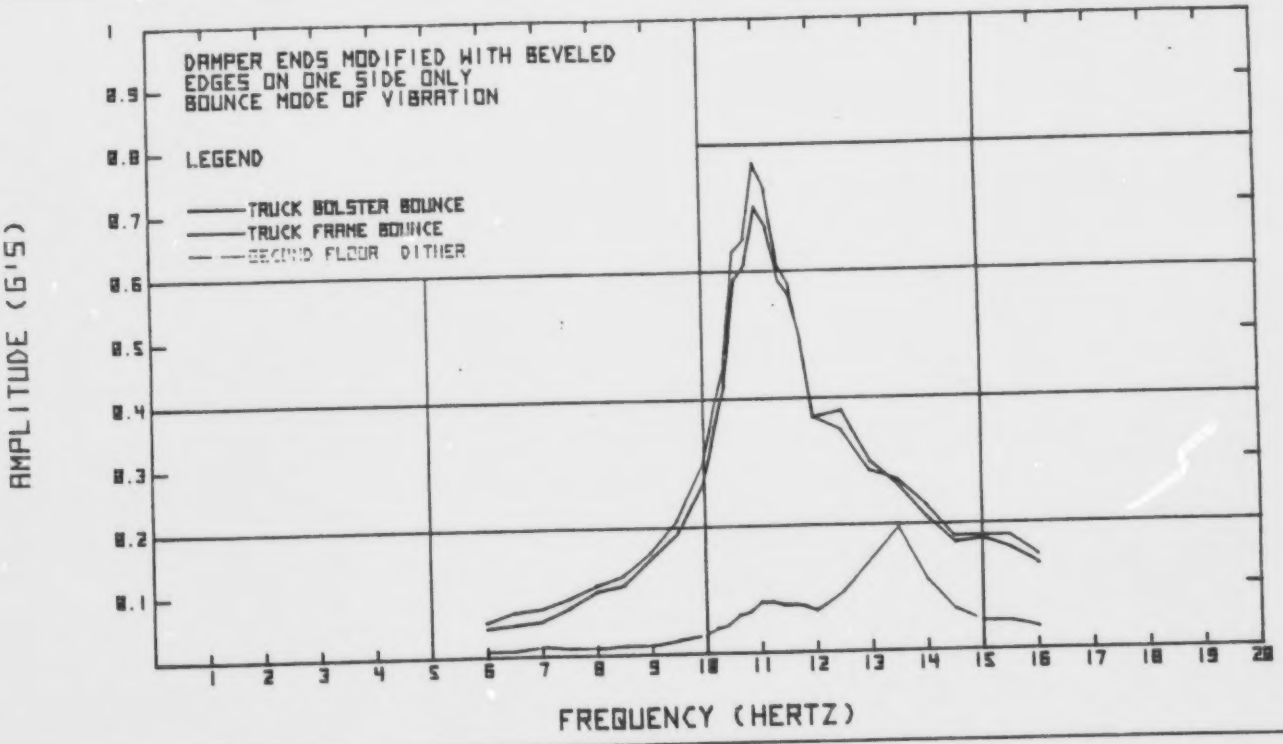
GOp0008m , GO02d02, Transfer Function between Upper & Lower Deck

APPENDIX C

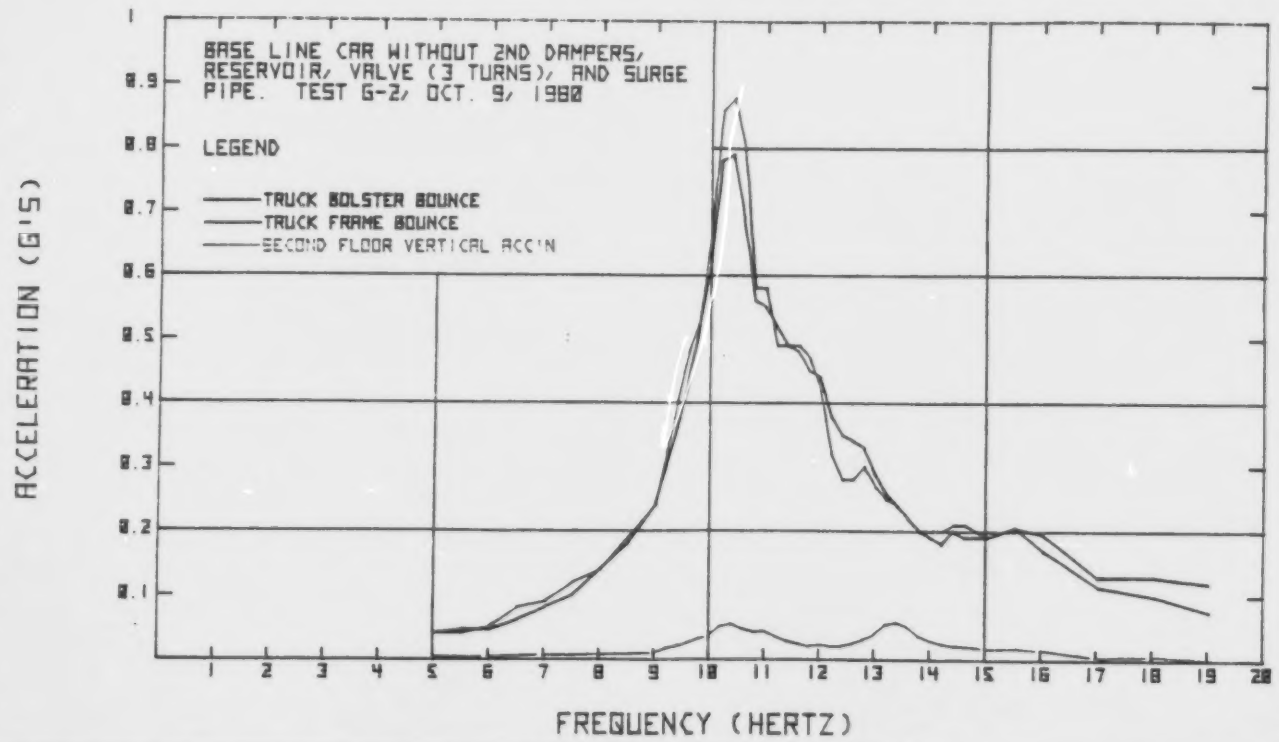
DOFASCO TRUCK RESONANCE TESTS

SOURCE: BILEVEL I DESIGN VERIFICATION TESTS

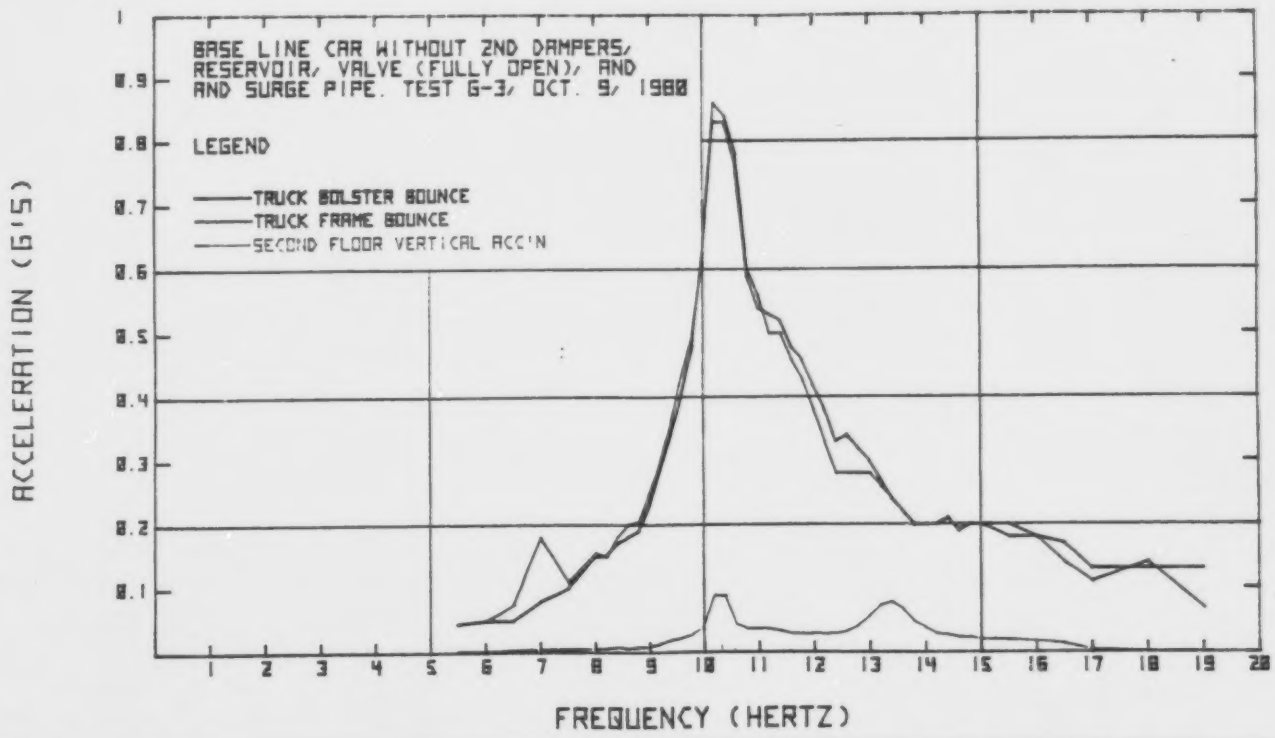
BI-LEVEL DITHER INVESTIGATION



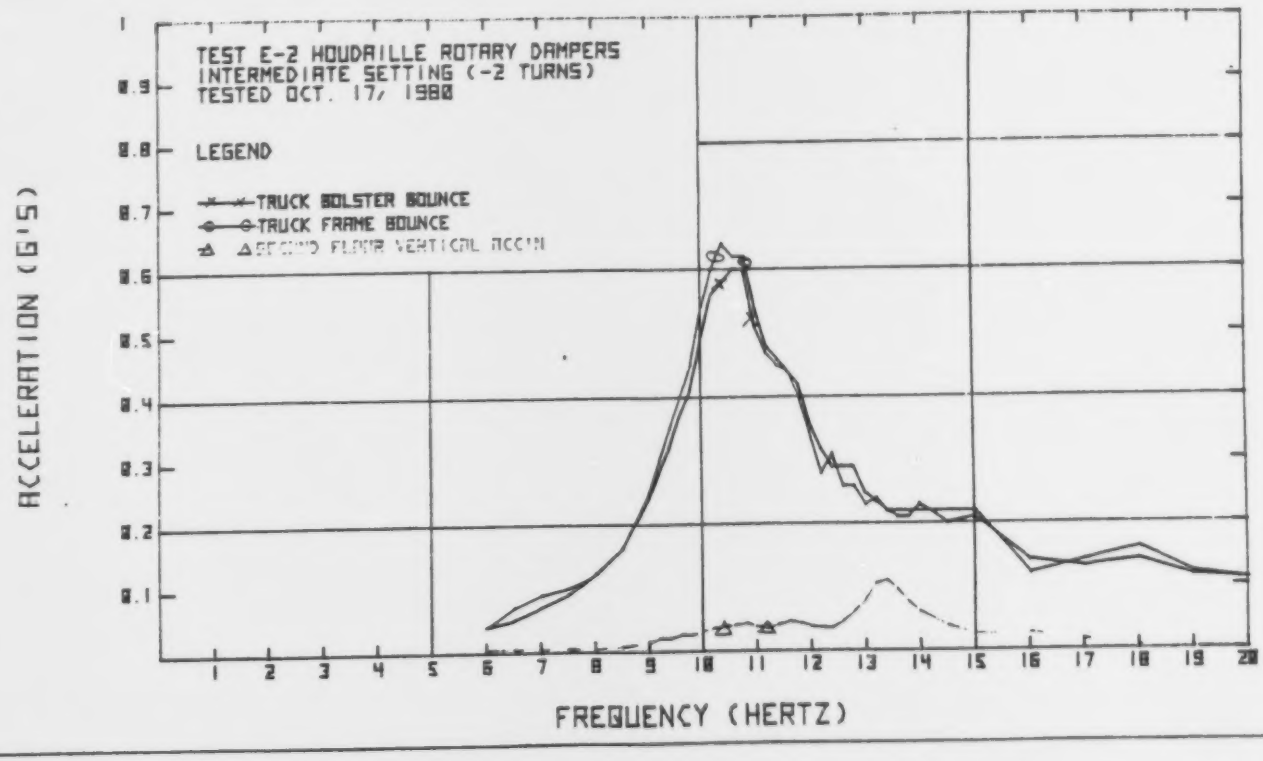
BI-LEVEL DITHER INVESTIGATION

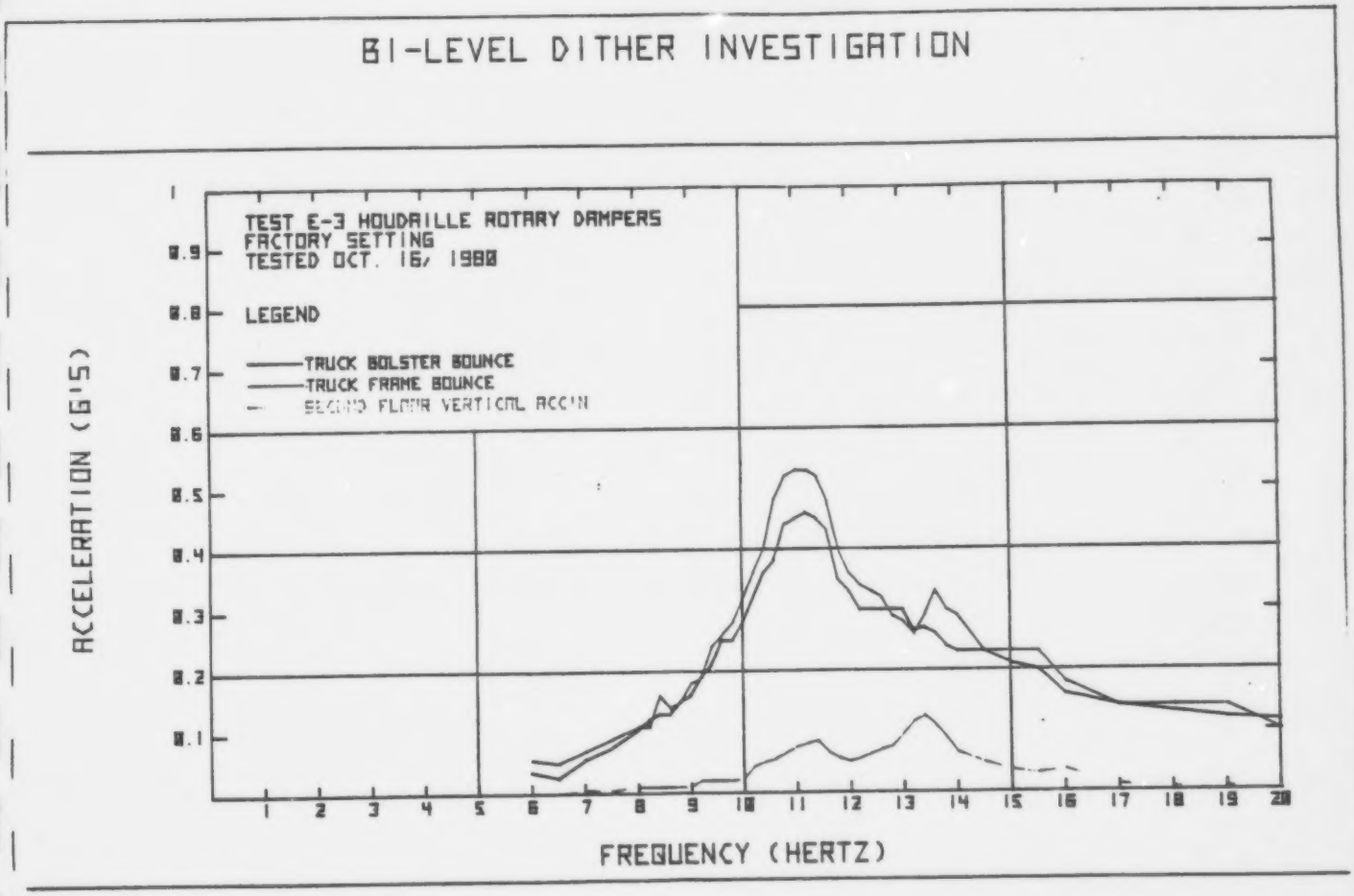

BASE LINE CAR WITHOUT 2ND DAMPERS, RESERVOIR, VALVE (3 TURNS), AND SURGE PIPE. TEST G-2, OCT. 9, 1980
LEGEND
TRUCK BOLSTER BOUNCE
TRUCK FRAME BOUNCE
SECOND FLOOR VERTICAL ACCIN
ACCELERATION (G'S)
FREQUENCY (HERTZ)

BI-LEVEL DITHER INVESTIGATION

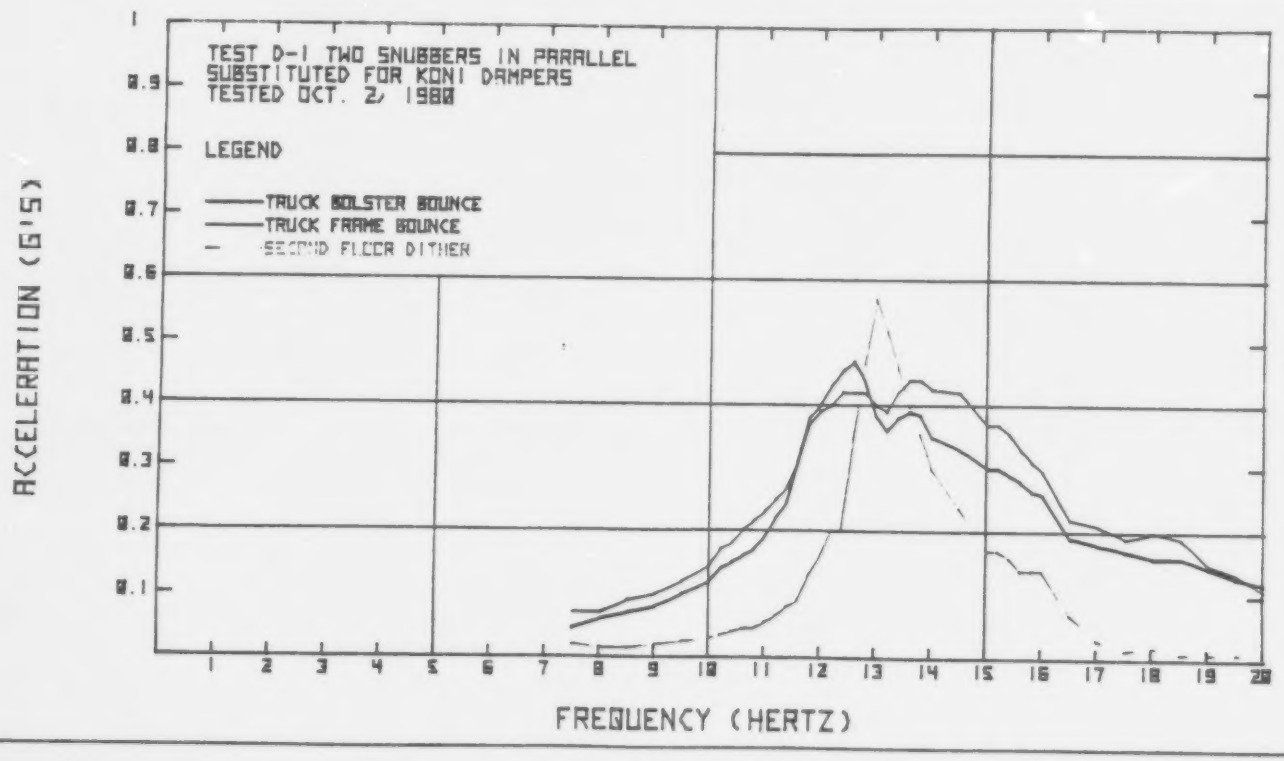

BASE LINE CAR WITHOUT 2ND DAMPERS, RESERVOIR, VALVE (FULLY OPEN), AND AND SURGE PIPE. TEST G-3, OCT. 9, 1980
LEGEND
TRUCK BOLSTER BOUNCE
TRUCK FRAME BOUNCE
SECOND FLOOR VERTICAL ACC'N
ACCELERATION (G'S)
FREQUENCY (HERTZ)

BI-LEVEL DITHER INVESTIGATION

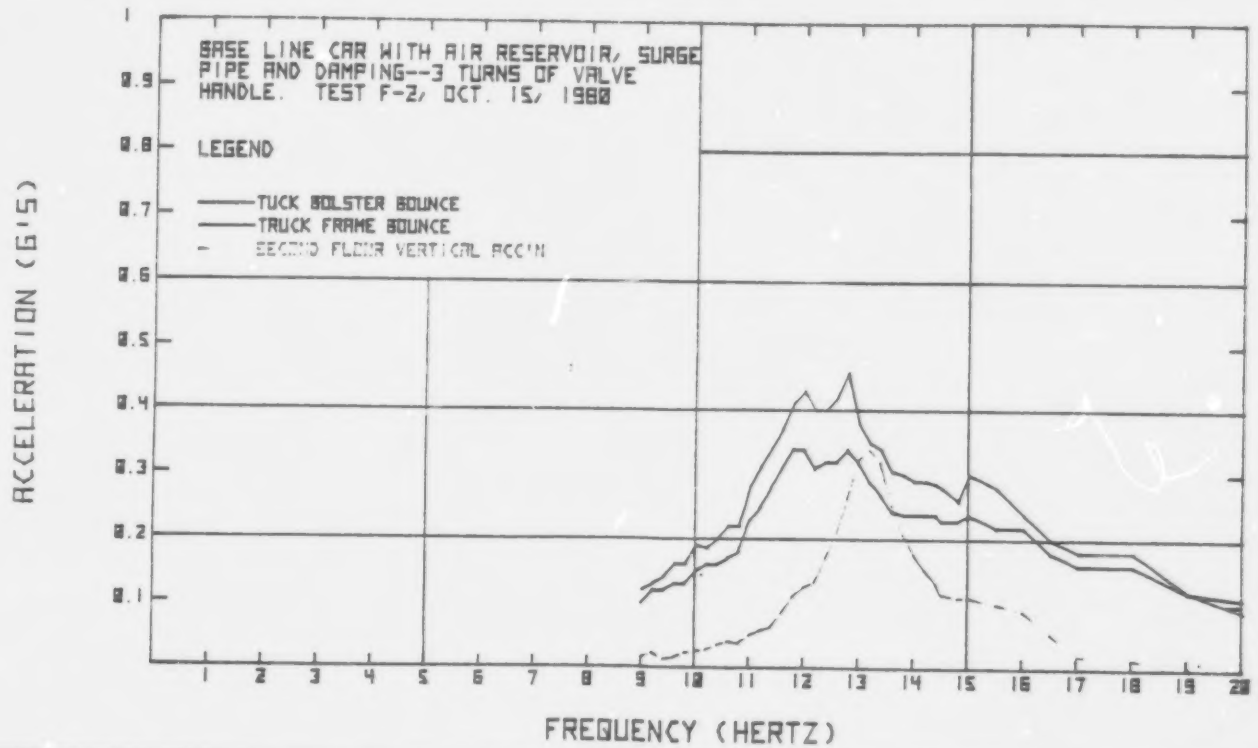



BI-LEVEL DITHER INVESTIGATION
TEST E-3 HOUDAILLE ROTARY DAMPERS
FACTORY SETTING
TESTED OCT. 16, 1980
LEGEND
TRUCK BOLSTER BOUNCE
TRUCK FRAME BOUNCE
SECOND FLOOR VERTICAL ACCN
ACCELERATION (G'S)
FREQUENCY (HERTZ)

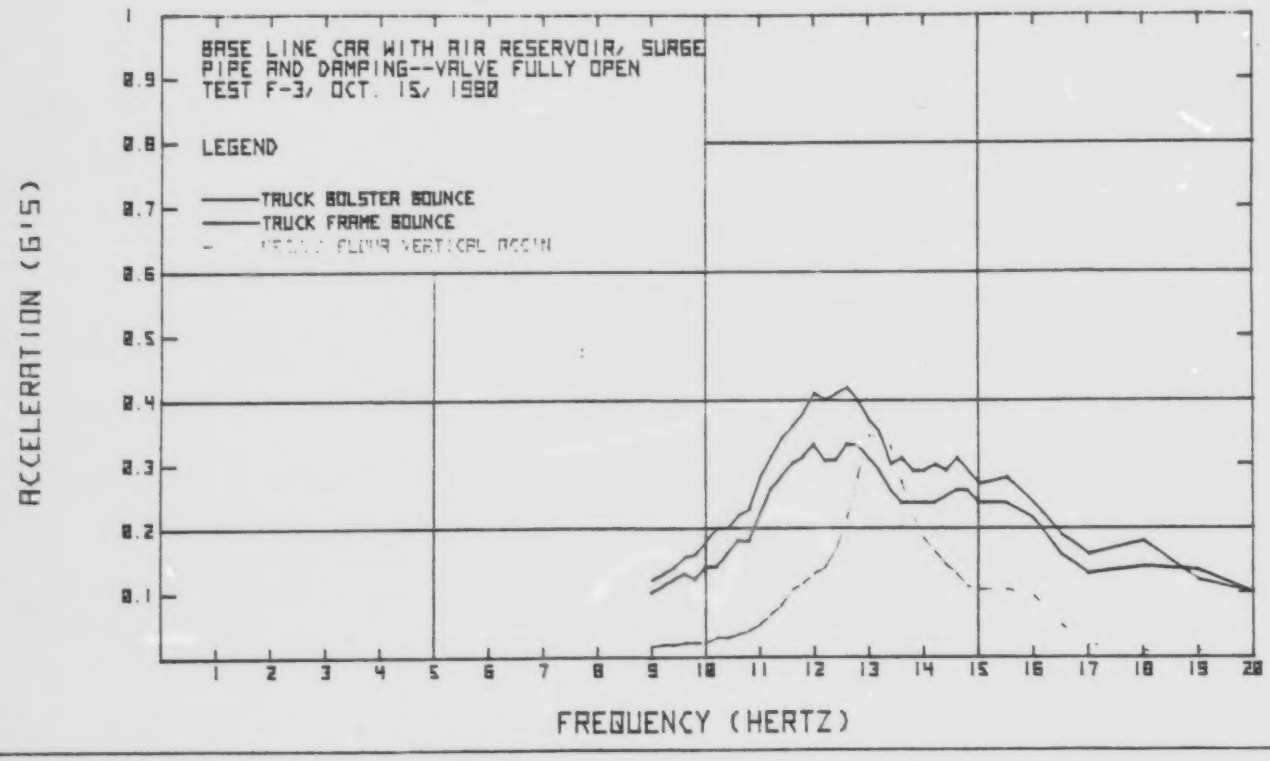
BI-LEVEL DITHER INVESTIGATION



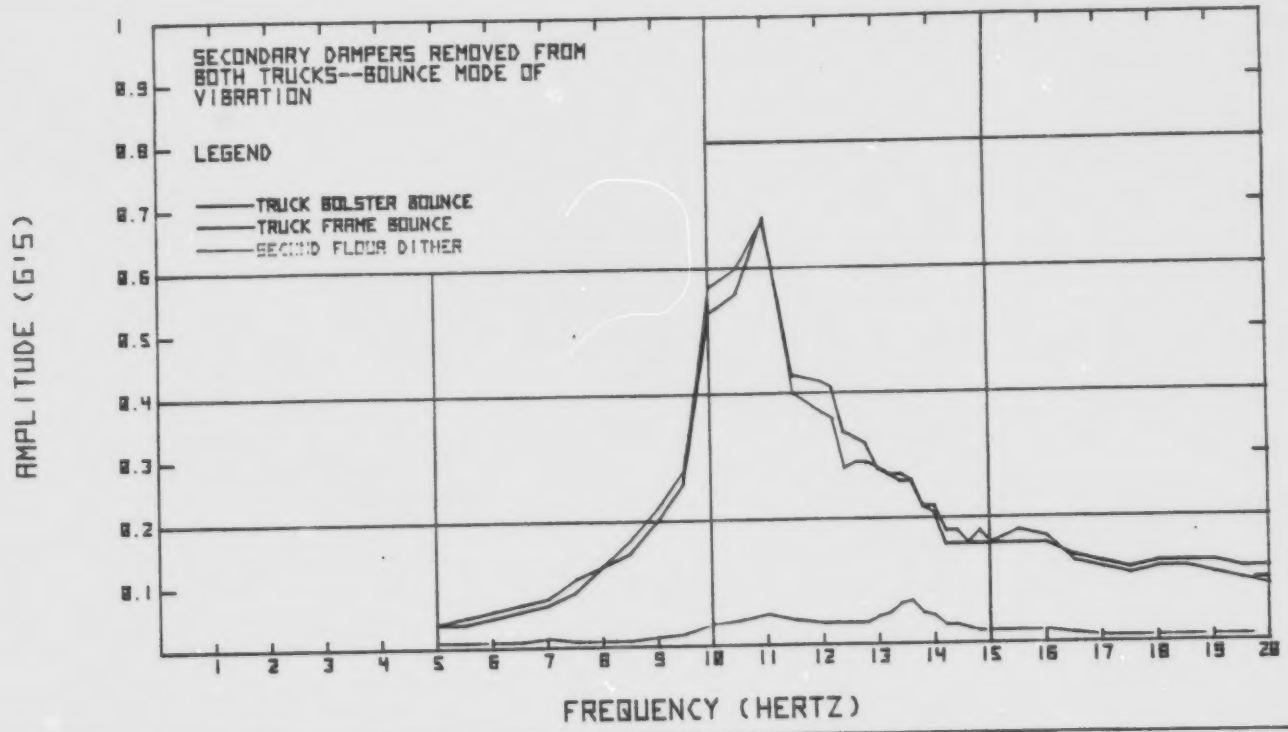
BI-LEVEL DITHER INVESTIGATION



BI-LEVEL DITHER INVESTIGATION


BASE LINE CAR WITH AIR RESERVOIR, SURGE PIPE AND DAMPING--VALVE FULLY OPEN
TEST F-3, OCT. 15, 1980
LEGEND
TRUCK BOLSTER BOUNCE
TRUCK FRAME BOUNCE
ACCELERATION (G'S)
FREQUENCY (HERTZ)

BI-LEVEL DITHER INVESTIGATION



APPENDIX D

IMPROVED DEFINITION TEST DATA

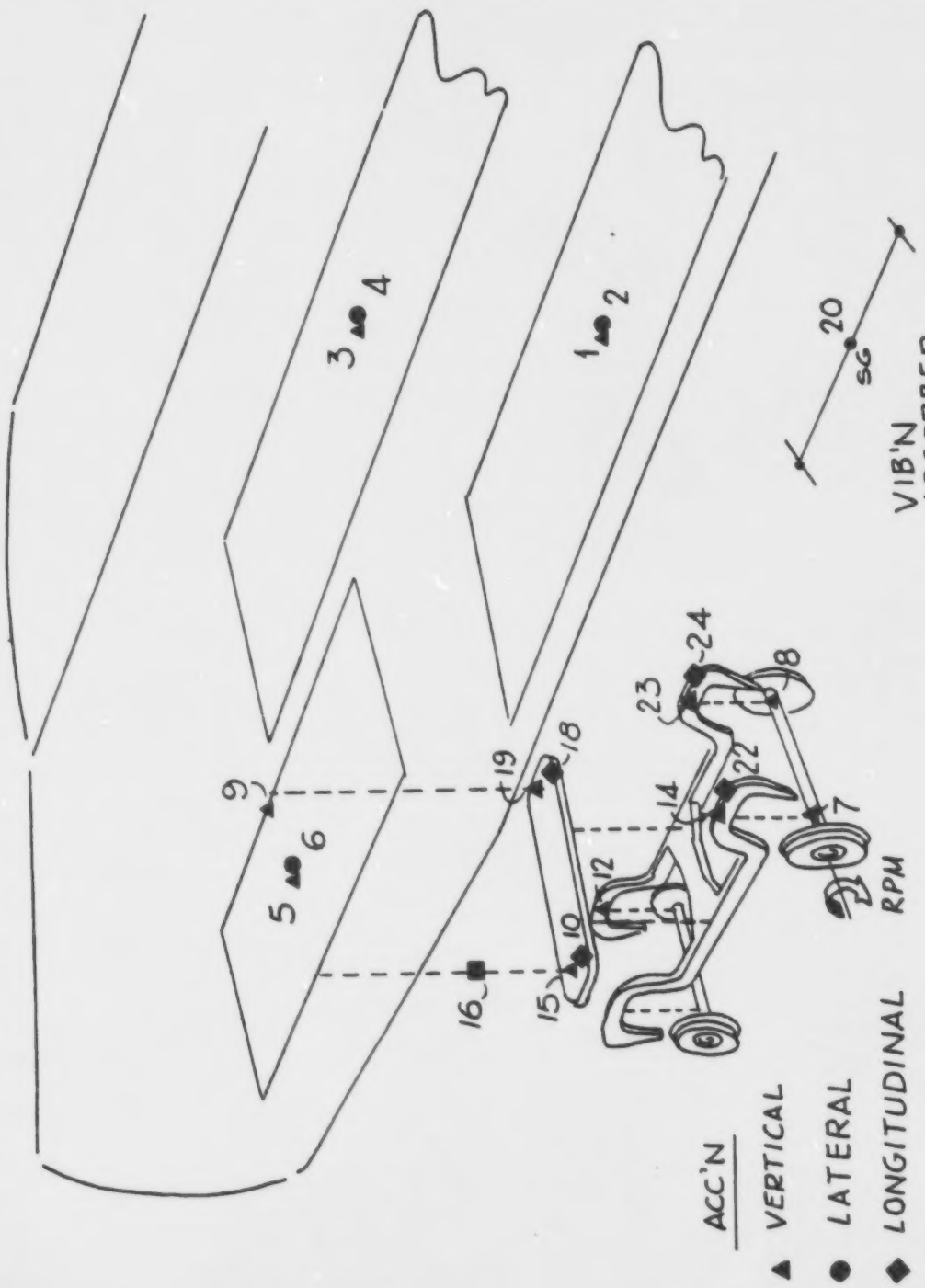
USED TO REFINE BASELINE MODEL

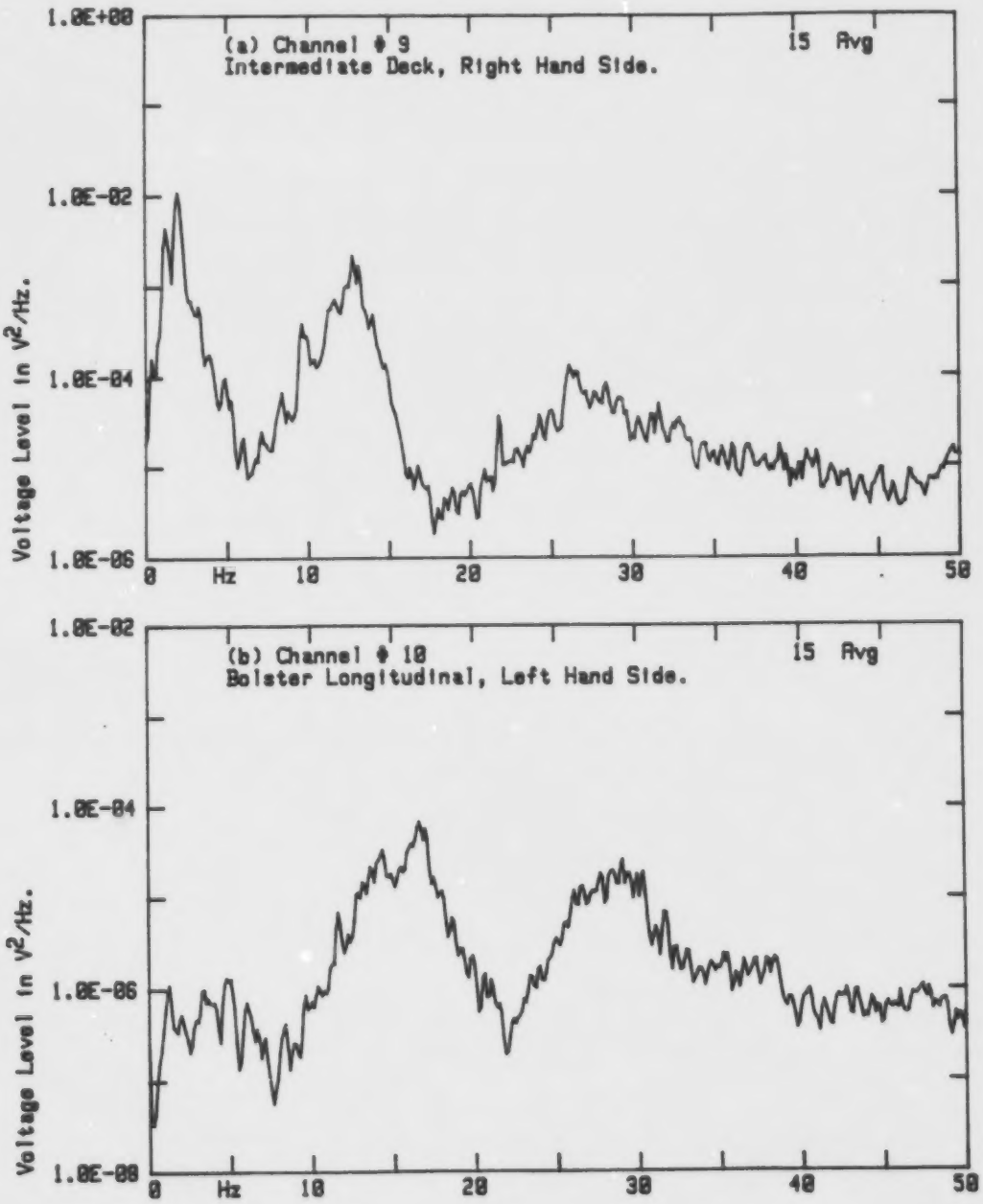
GO Transit Bi_level testing 1988.06.16

Channel Configuration:

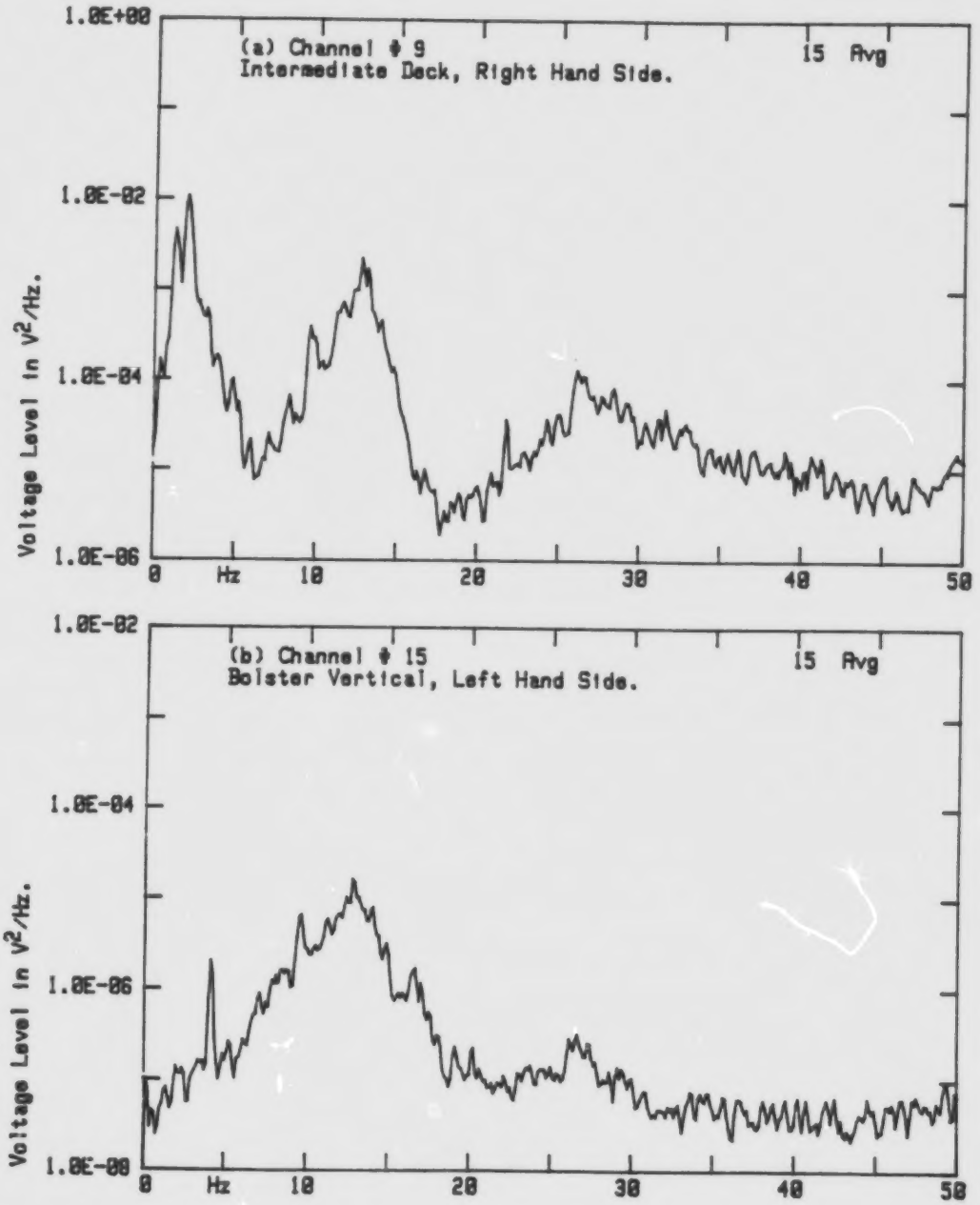
Channel 1:	Vertical acceleration on centre of the lower deck.		
	Gain 0.228,	Phase -62.5,	230.83E-03 V/ms^-2
Channel 2:	Lateral acceleration on centre of the lower deck.		
	Gain 0.226,	Phase -63.0	228.80E-03 V/ms^-2
Channel 3:	Vertical acceleration on centre of the upper deck.		
	Gain 0.230,	Phase -62.7	227.18E-03 v/ms^-2
Channel 4:	Lateral acceleration on centre of the upper deck.		
	Gain 0.226,	Phase -63.3	228.80E-03 V/ms^-2
Channel 5:	Vertical acceleration on centre of the interm. deck		
	Gain 0.396,	Phase -57.3	400.91E-03 V/ms^-2
Channel 6:	Lateral acceleration on centre of the interm. deck.		
	Gain 0.479,	Phase -56.9	484.94E-03 V/ms^-2
Channel 7:	Vertical acceleration of the left axle box.		
	Gain 4.404,	Phase 22.8	4.4128E-03 V/ms^-2
Channel 8:	Vertical acceleration of the right axle box.		
	Gain 3.786,	Phase 23.6	3.7519E-03 V/ms^-2
Channel 9:	Vertical acceleration of the side of interm. deck.		
	Gain 21.05,	Phase 90.5	210.50E-03 V/ms^-2
Channel 10:	Longitudinal acceleration of left bolster		
	Gain 0.720,	Phase -65.4	3.4416E-03 V/ms^-2
Channel 11:	n/a		
Channel 12:	Vertical acceleration of right frame, forward.		
	Gain 1.066,	Phase -55.0	5.0209E-03 V/ms^-2
Channel 13:	n/a		
Channel 14:	Vertical acceleration of left frame.		
	Gain 4.242,	Phase 125.9	4.2971E-03 V/ms^-2
Channel 15:	Vertical acceleration of left bolster.		
	Gain 2.496,	Phase 122.2	1.9544E-03 V/ms^-2
Channel 16:	Pressure in Airbag. (left)		
	Gain ,	Phase	
Channel 17:	n/a		
Channel 18:	Longitudinal acceleration of right bolster.		
	Gain 2.168,	Phase -54.6	10.233E-03 V/ms^-2
Channel 19:	Vertical acceleration of right bolster.		
	Gain 3.633,	Phase -62.3	3.0081E-03 V/ms^-2
Channel 20:	Strain gauge on vibration absorber.		
	Gain ,	Phase	
Channel 21:	n/a		
Channel 22:	Longitudinal acceleration of left frame.		
	Gain 4.416,	Phase 125.5	4.4822E-03 V/ms^-2
Channel 23:	Vertical acceleration of right frame.		
	Gain 4.426,	Phase -56.3	4.4880E-03 V/ms^-2
Channel 24:	Longitudinal acceleration of right frame.		
	Gain 4.397,	Phase -54.4	4.4717E-03 V/ms^-2

All sensitivities etc. at 50 Hz.

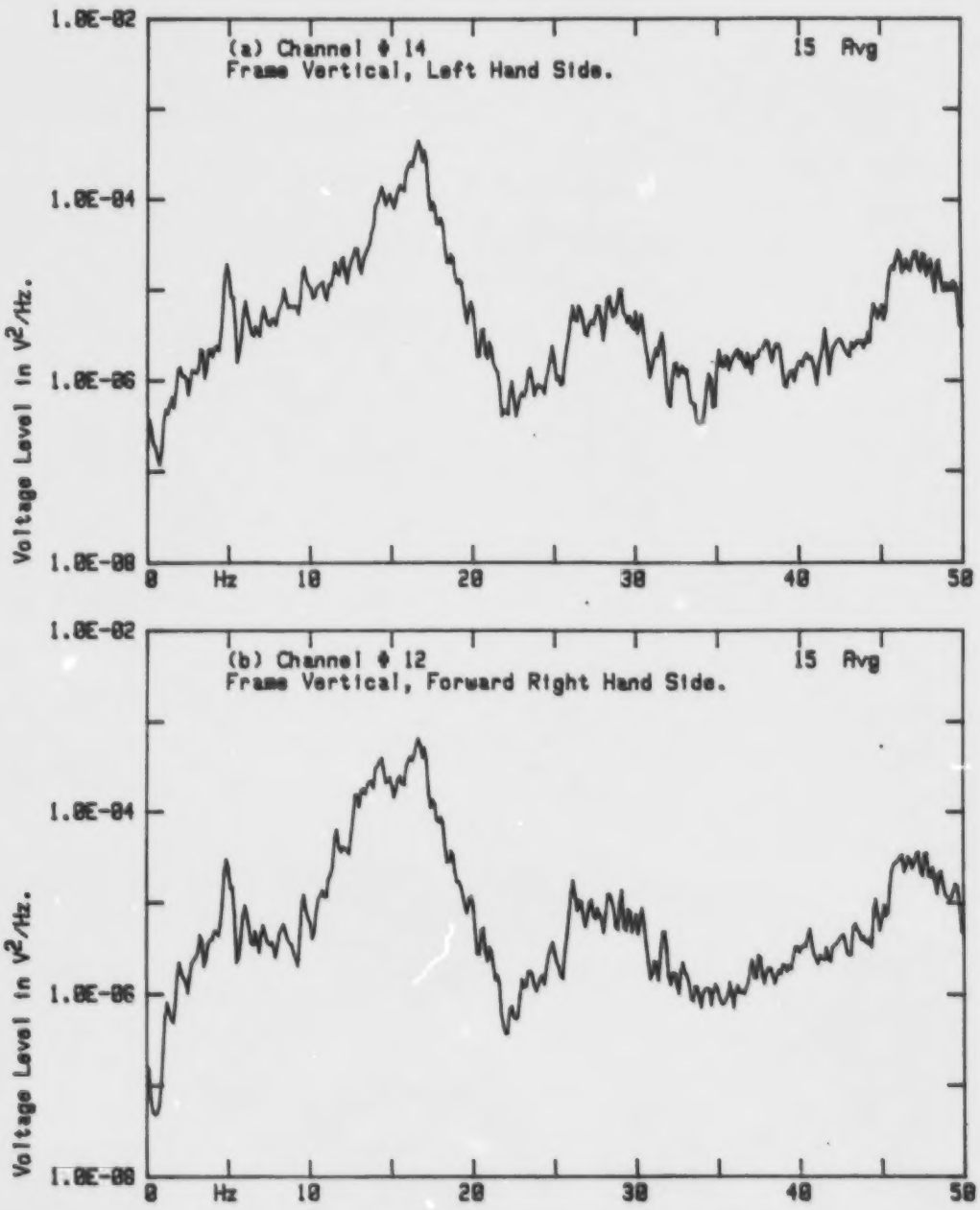

BILEVEL CAR 2030 INSTRUMENTATION
ACC'N
▲ VERTICAL
● LATERAL
◆ LONGITUDINAL
RPM
VIB'N ABSORBER
SG
1
2
3
4
5
6
7
8
9
10
12
14
15
16
18
19
20
22
23
24



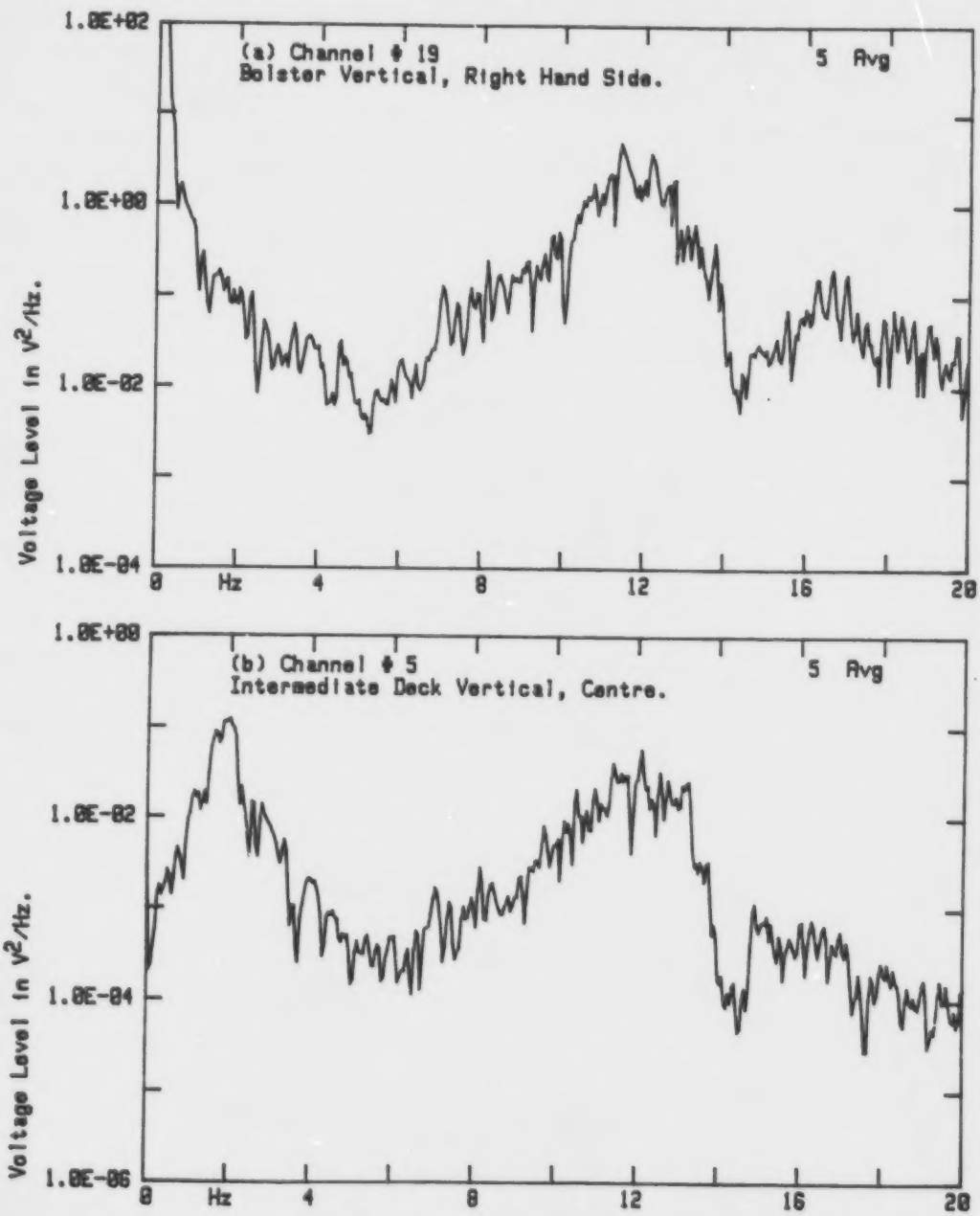
GOp00k1a, GOk4142_01 GO Transit Testing 1988.06.



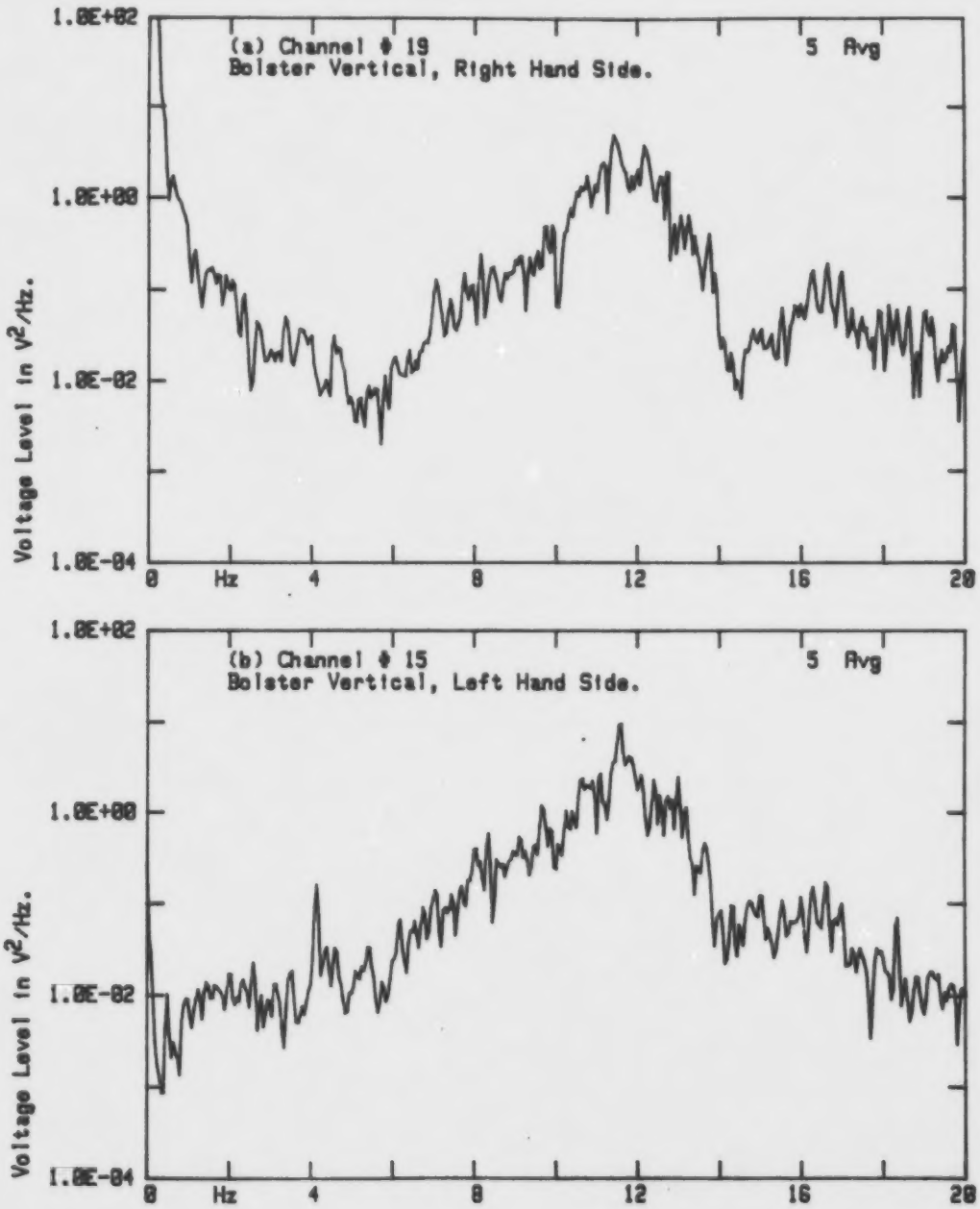
GOp00k1b, GOk4147_01 GO Transit Testing 1988.06.



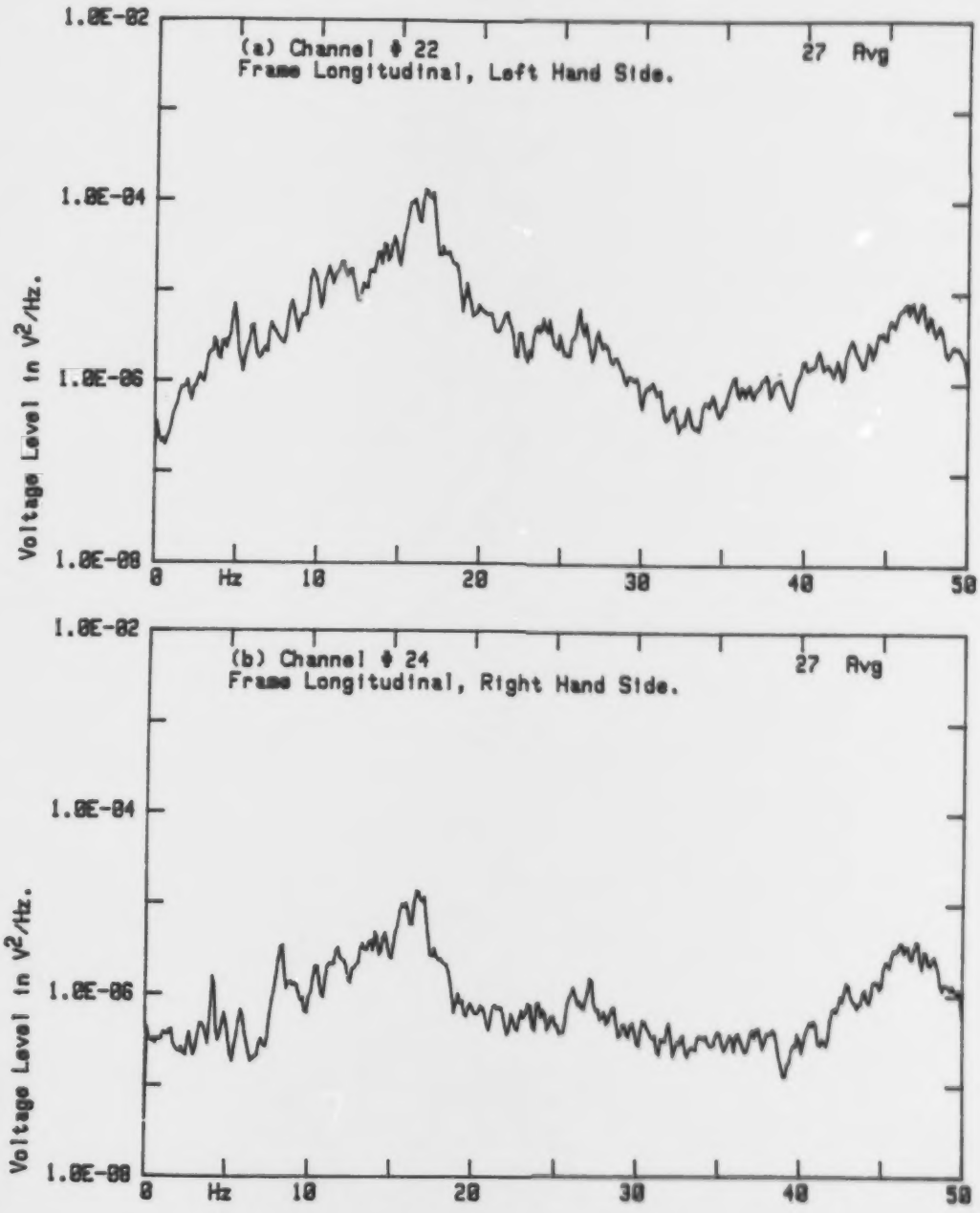
GOp00k1c, GOk2624_01 GO bilevel testing 1988-06-21



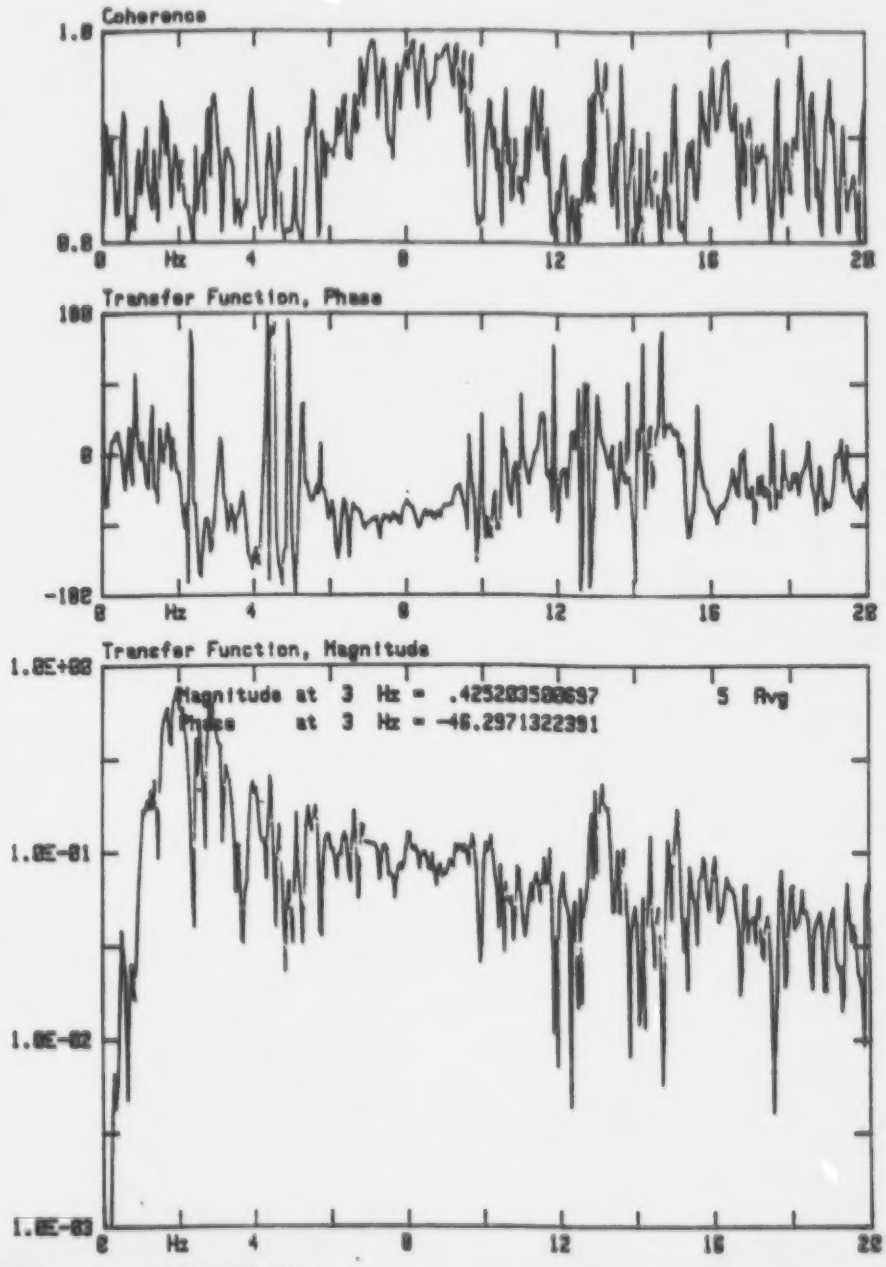
GOp00k1e, GOk1905_01 GO Transit Testing 1988.06.21.



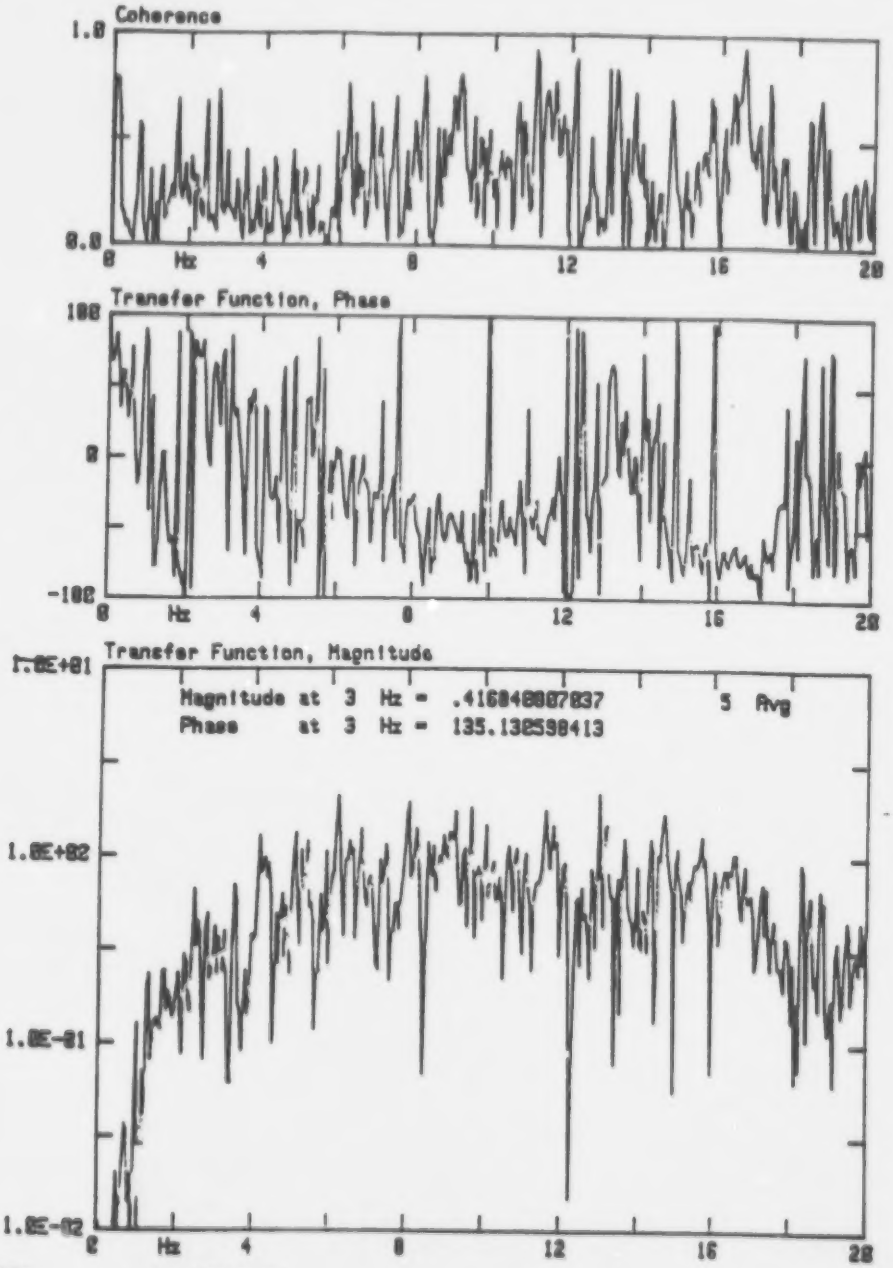
GOp00k1f, GOk1915_01 GO Transit Testing 1988.06.21.



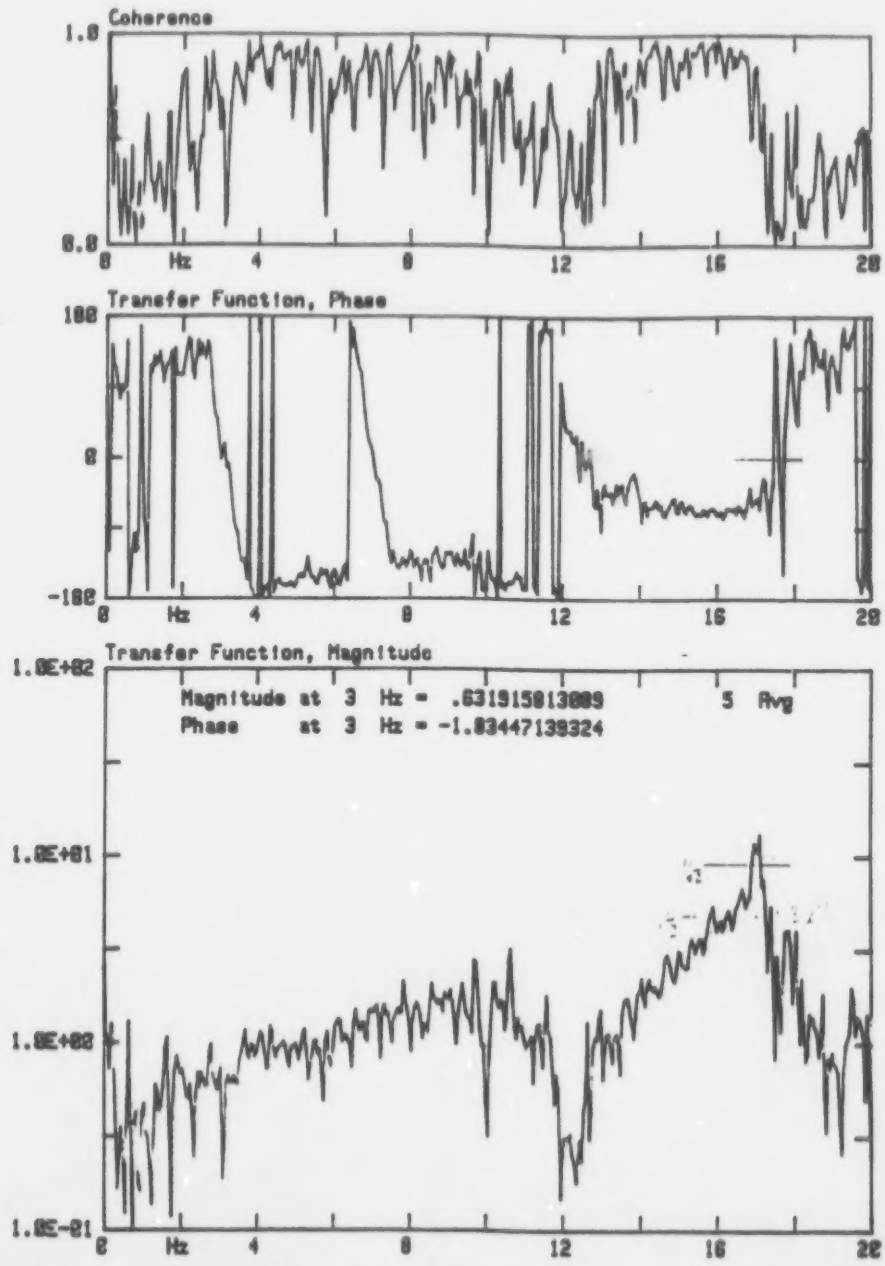
GOp00k1d, GOk6668_01 GO Transit Testing 1988.06.21



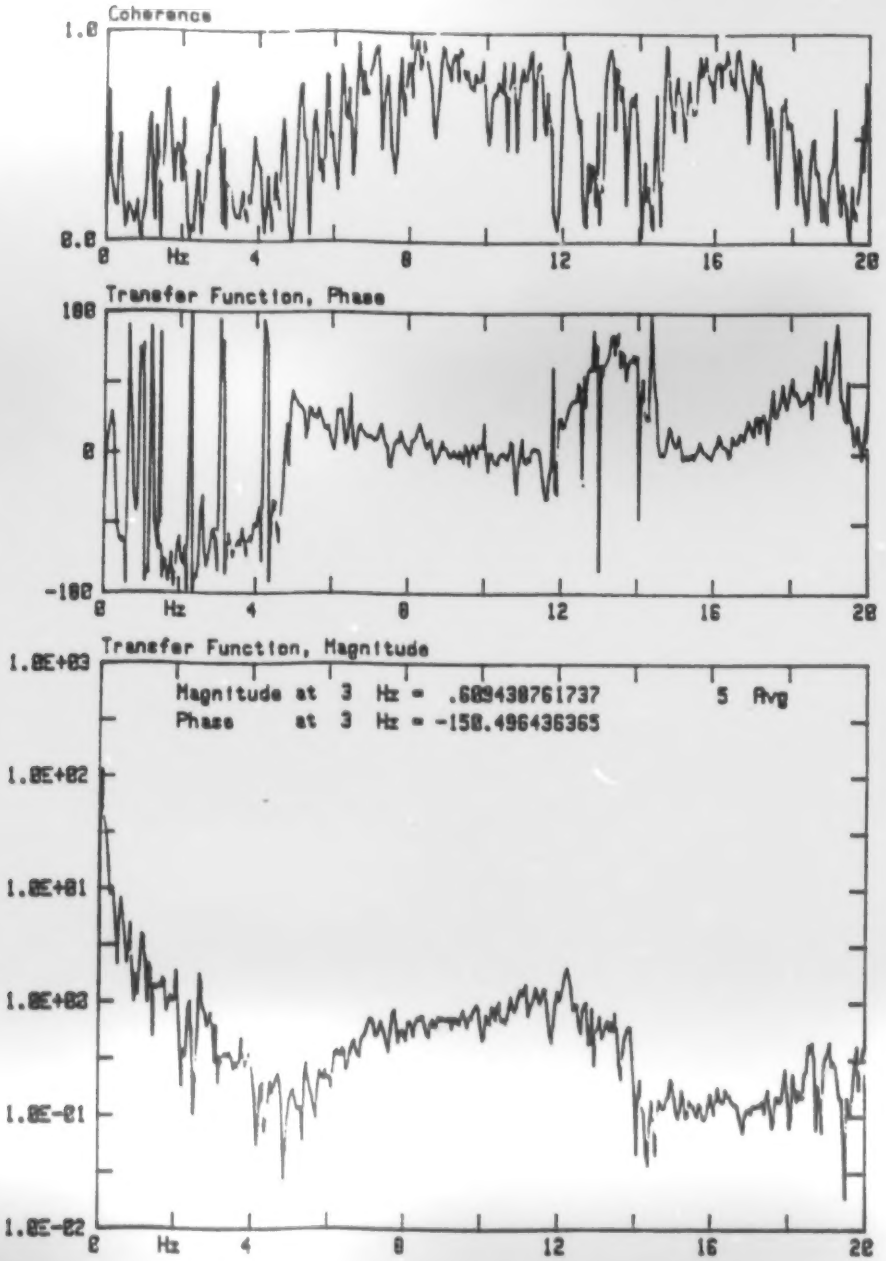
GG02 , G0%1905_01
Channel A is channel 19, Channel b is channel 05, Port Credit to Clarkson
Start 283:12:13:45, 60C revs out of station, Baseline Data.



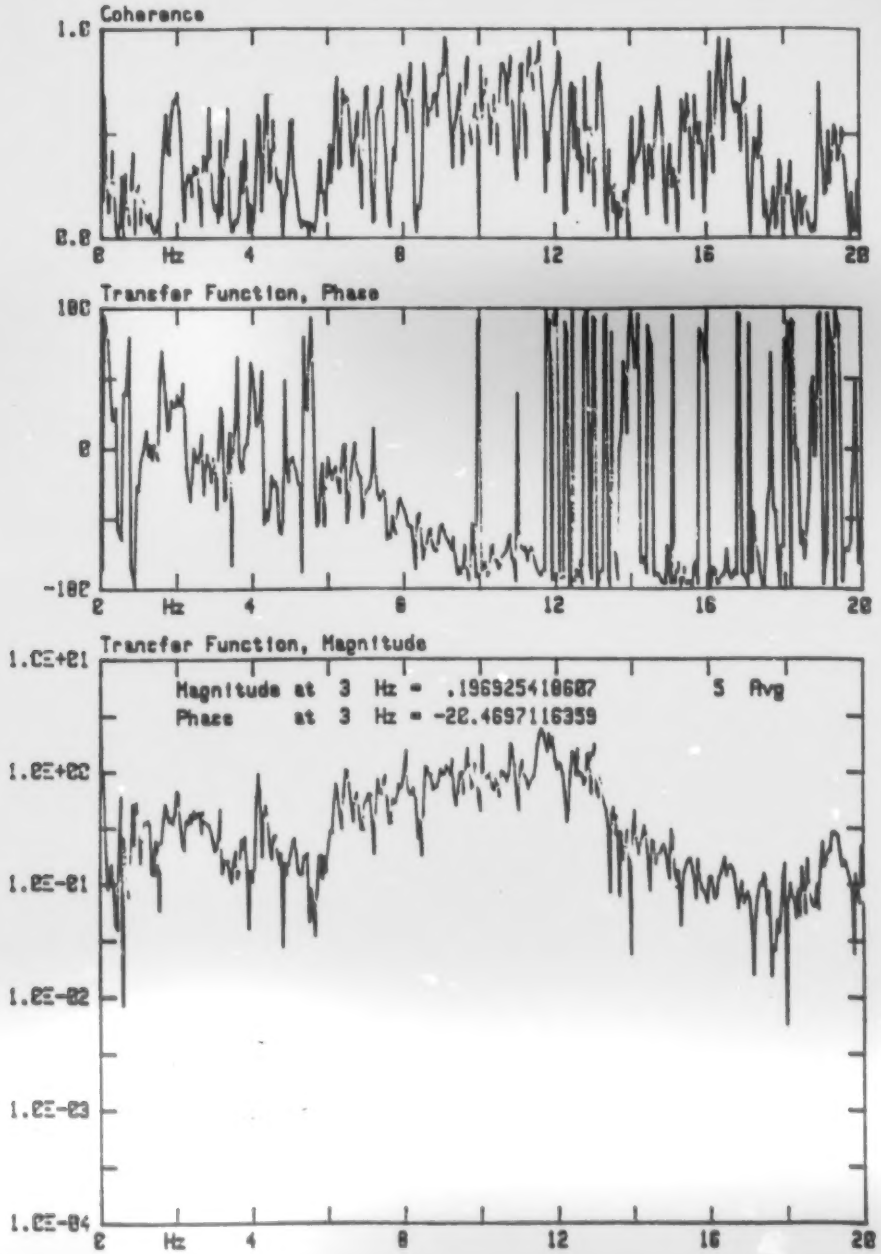
GG82 , GOk1915_01
Channel A is channel 18, Channel b is channel 15, Port Credit to Clarkson
Start 283:18:13:45, 682 revs out of station, Baseline Data.



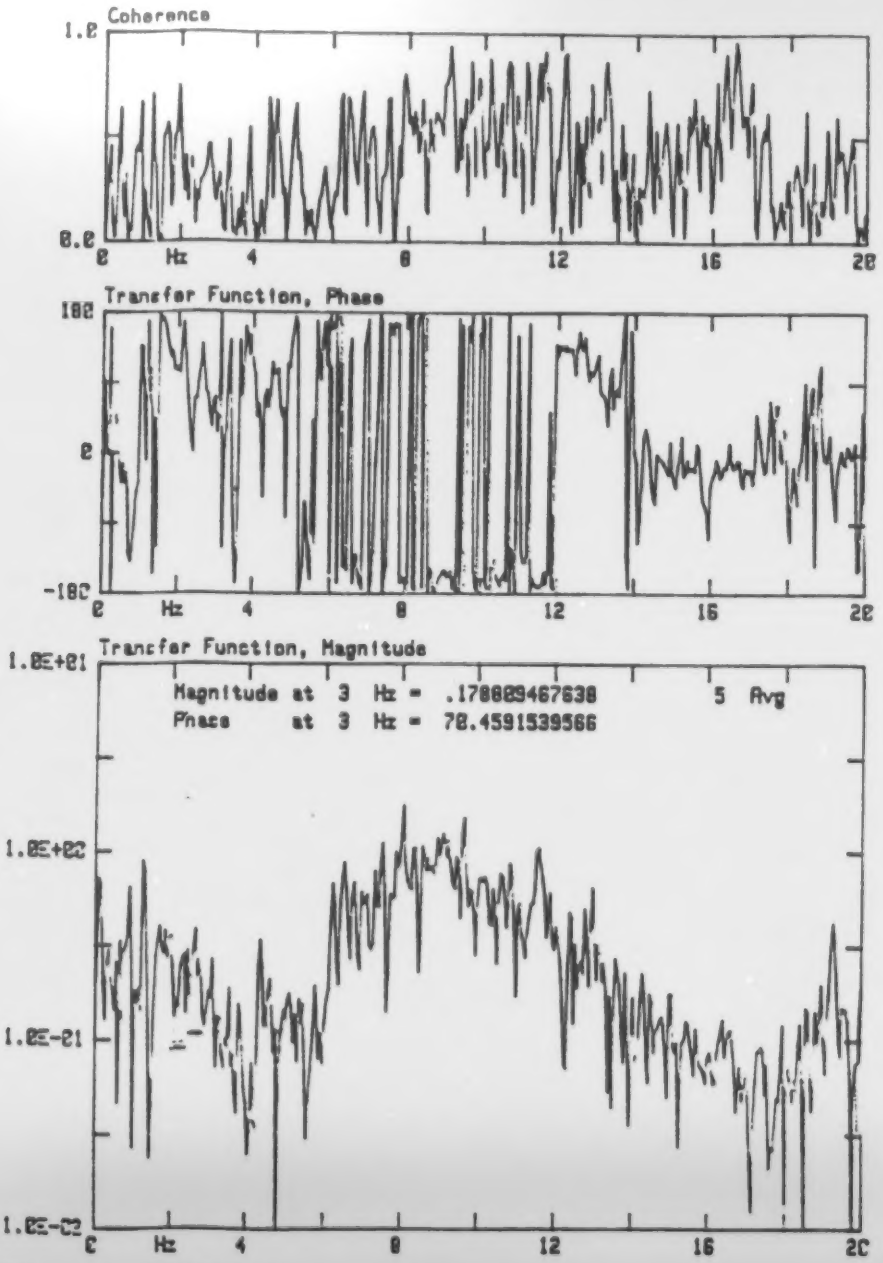
GG02 , GOk0714_01
Channel A is channel 07, Channel B is channel 14, Port Credit to Clarkson
Start 203:18:13:45, 602 revs out of station, Baseline Data.



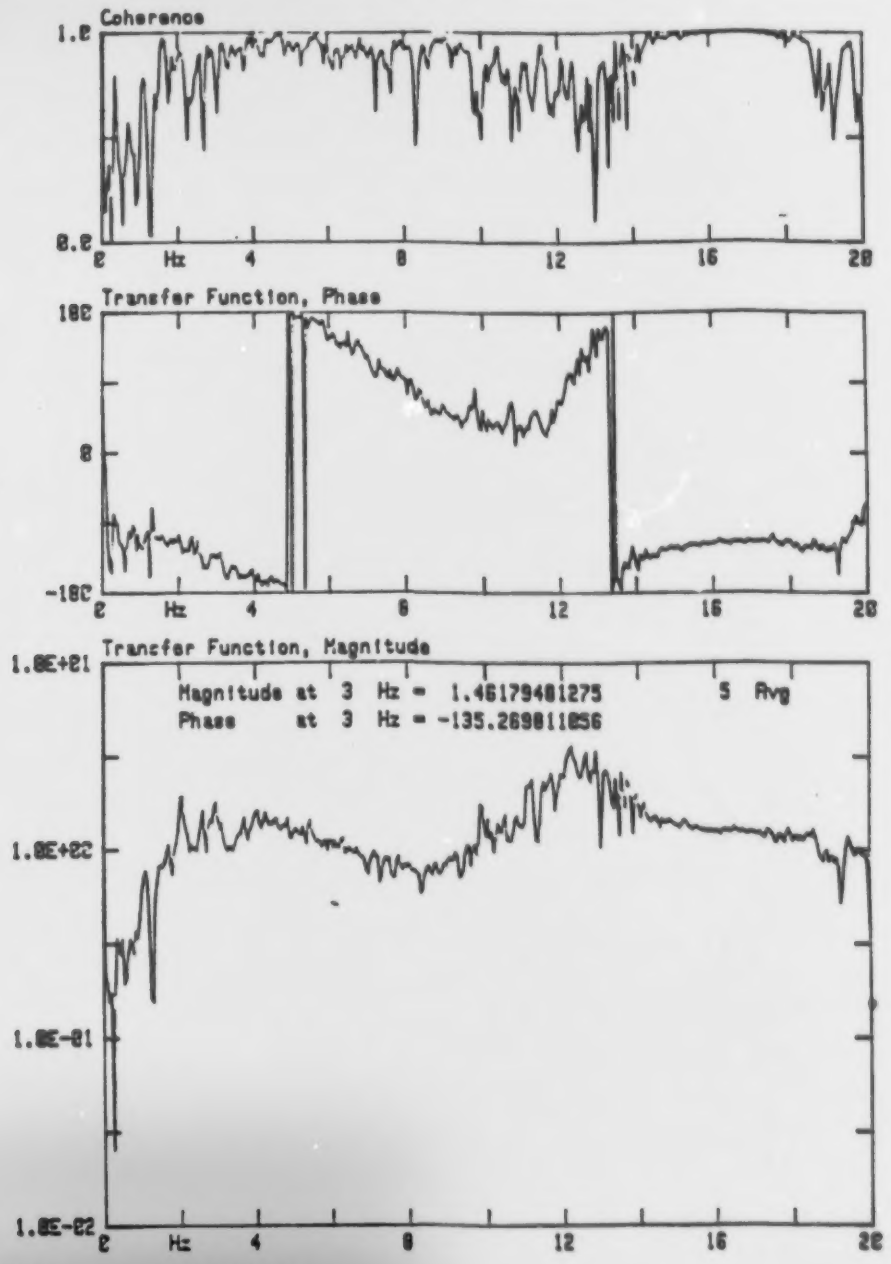
GG02 , GOk2319_0:
Channel A is channel 23, Channel b is channel 19, Port Credit to Clarkson
Start 23:10:13:45, GCC revs out of station, Baseline Data.



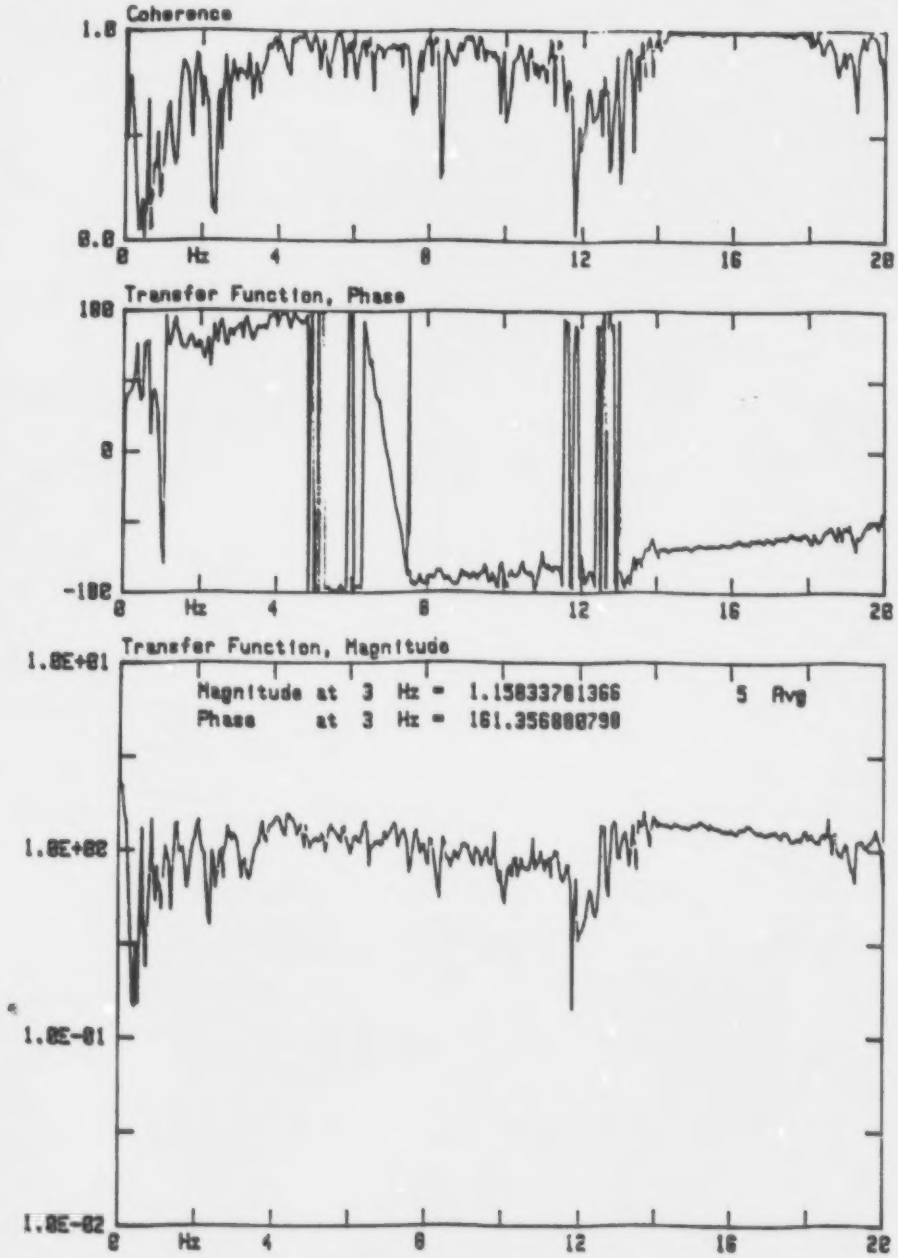
GG02 , GOk2315_01
Channel A to channel 23, Channel b to channel 15, Port Credit to Clarkson
Start 203:10:13:45, 600 revs out of station., Baseline Data.



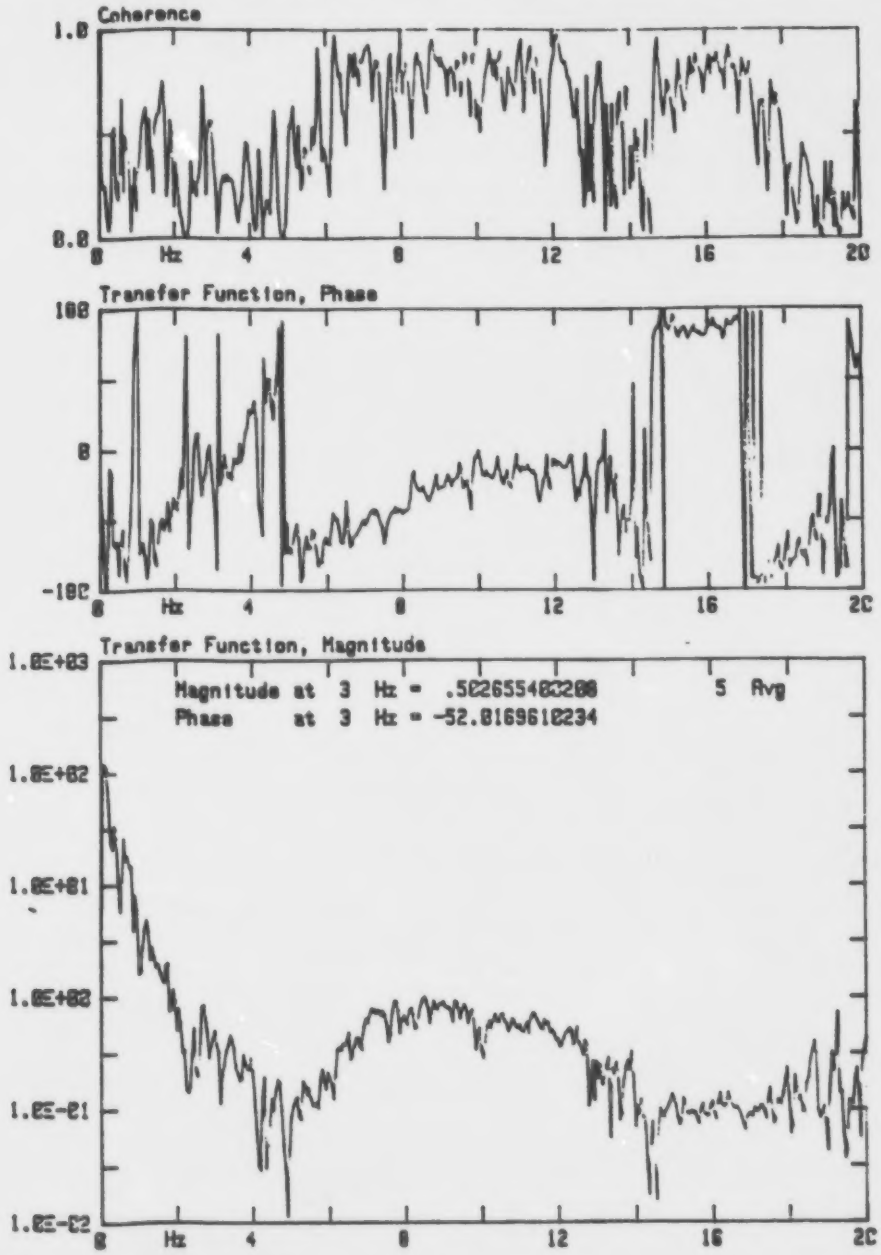
GGD2 , GOk1215_01
Channel A to channel 12, Channel B to channel 15, Port Credit to Clarkson
Start 203:10:13:45, 600 revs out of station, Baseline Data.



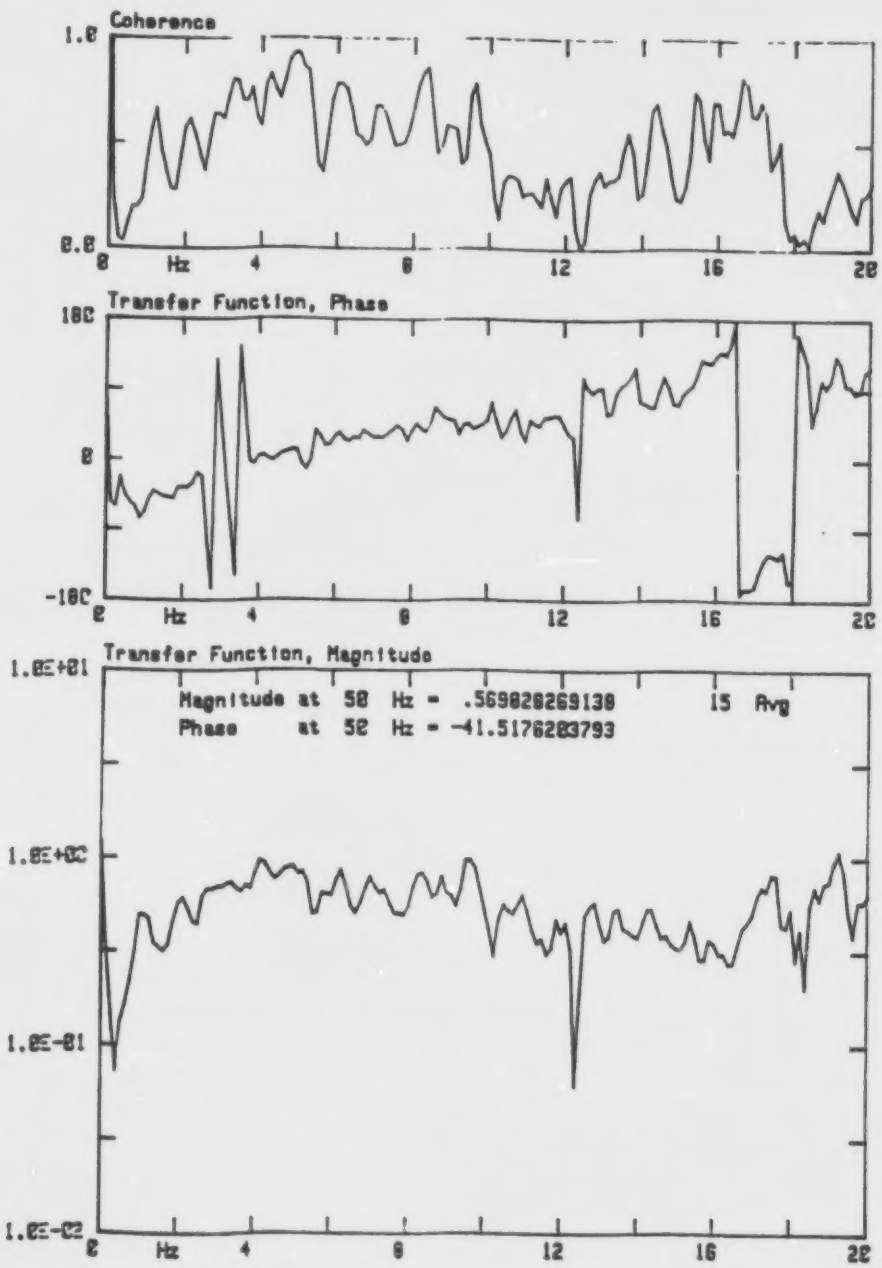
GG02 , GOk2312_01
Channel A is channel 23, Channel b is channel 12, Port Credit to Clarkson
Start 203:10:13:45, 600 revs out of station, Baseline Data.



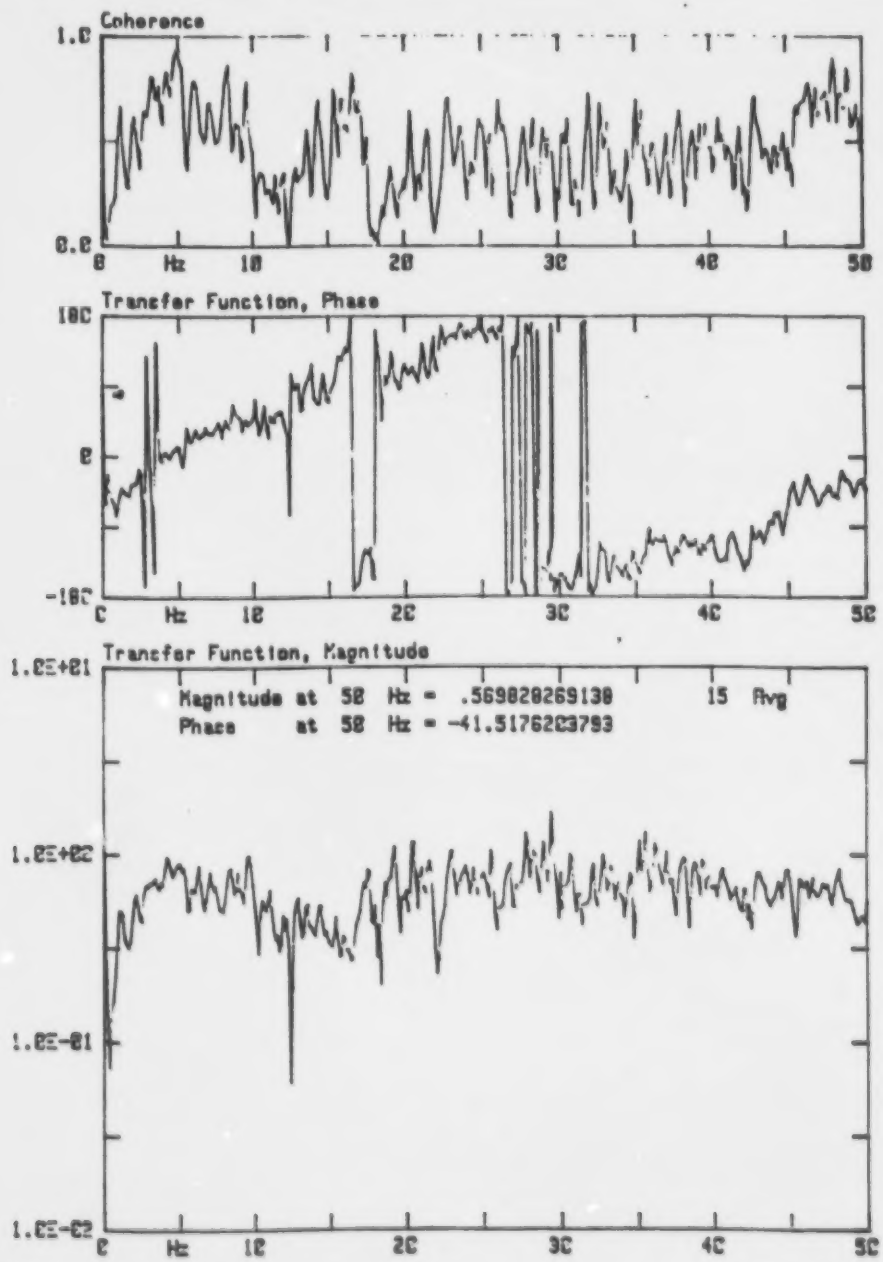
GG02 , GOk2314_01
Channel A is channel 23, Channel b is channel 14, Port Credit to Clarkson
Start 283:18:13:45, GEC revt out of station, Baseline Data.



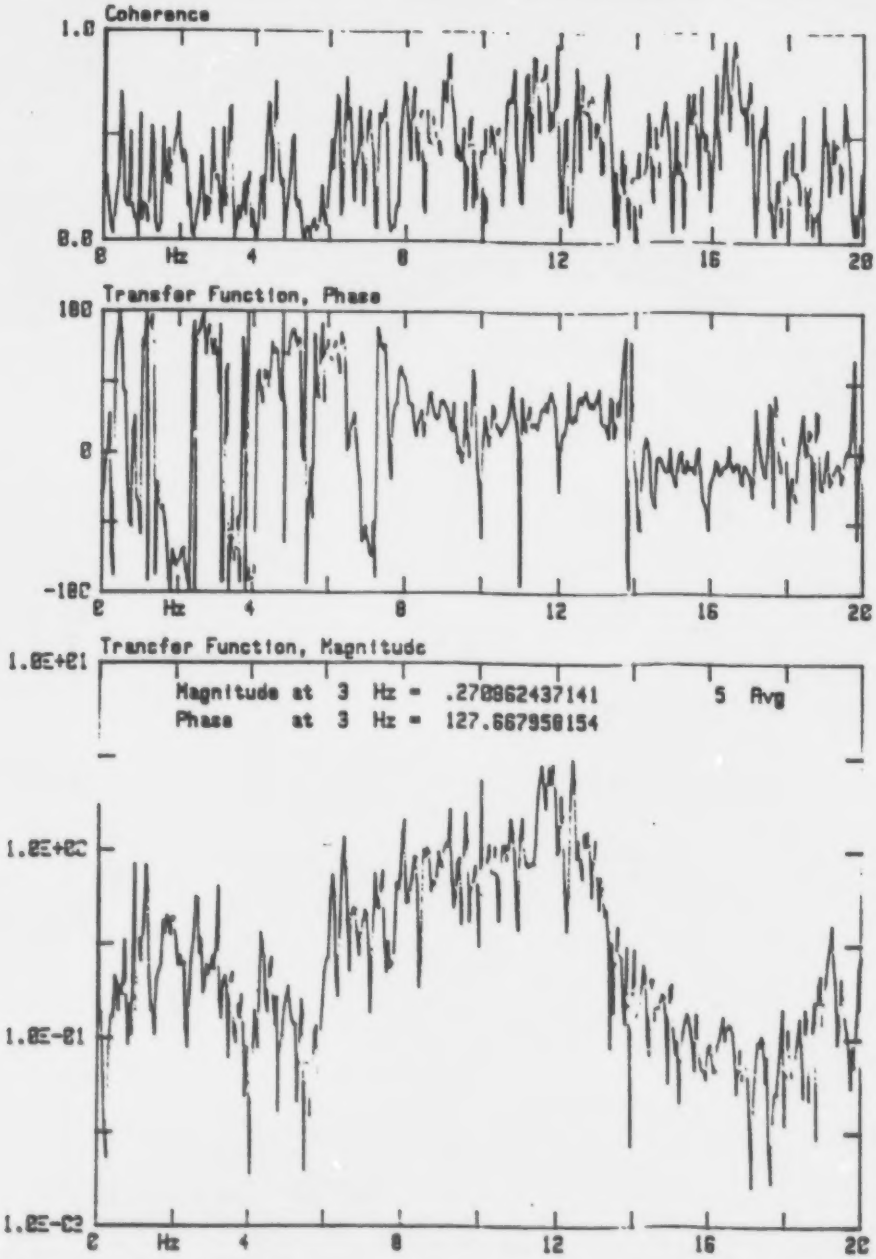
GGO2 , GOk1219_01
Channel A is channel 12, Channel B is channel 19, Port Credit to Clarkson
Start 203:10:13:45, 600 revs out of station, Baseline Data.



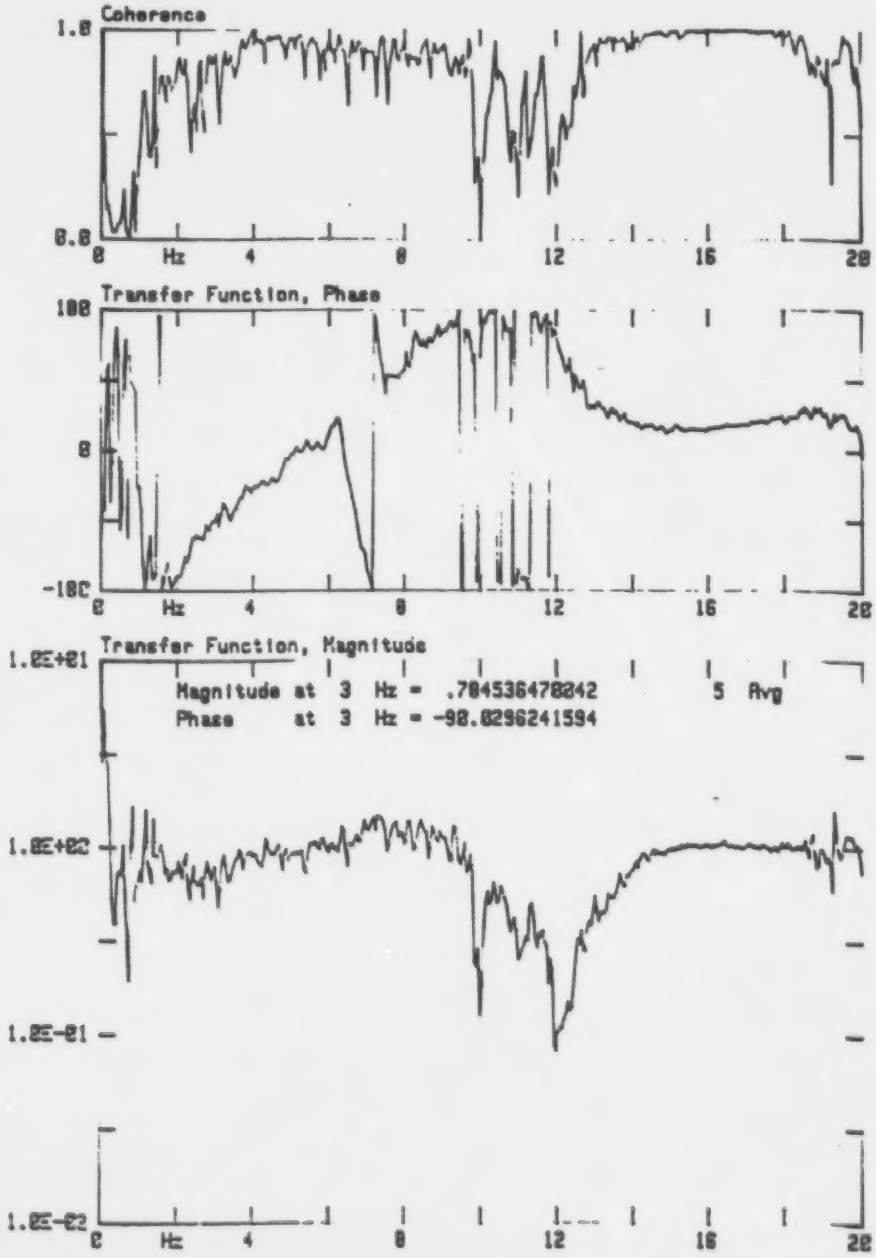
GGC2 , GOk8708_01
GO Transit Testing 1988.06.21
channel A: channel 7, channel B: channel 8, around 223.10.14.02



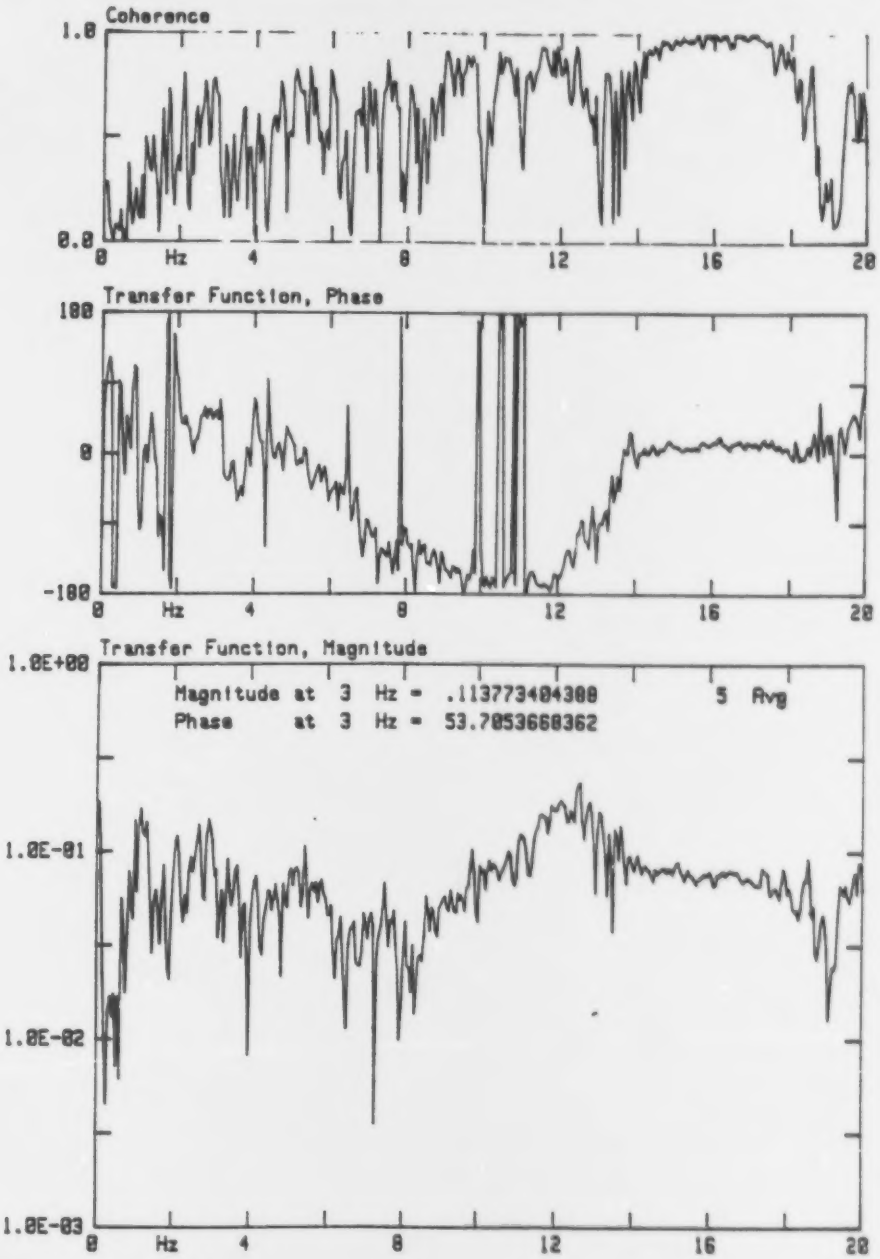
GGC2 , GOk0709_01
GO Transit Testing 1988.06.21
channel A: channel 7, channel B: channel 8, around 203.10.14.00



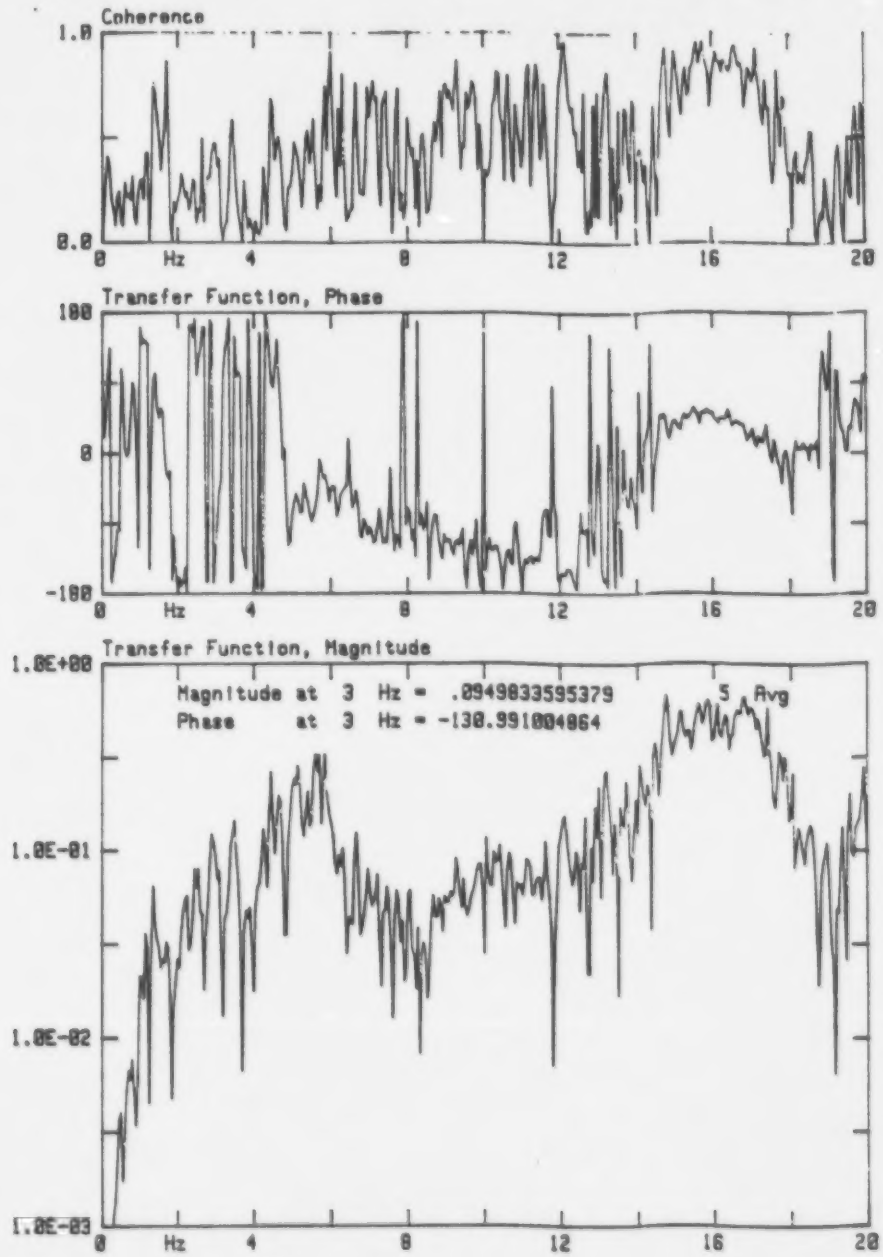
GG02 , GOk1415_01
Channel A is channel 14, Channel B is channel 15, Port Credit to Clarkson
Start 203:18:13:45, 600 revs out of station, Baseline Data.



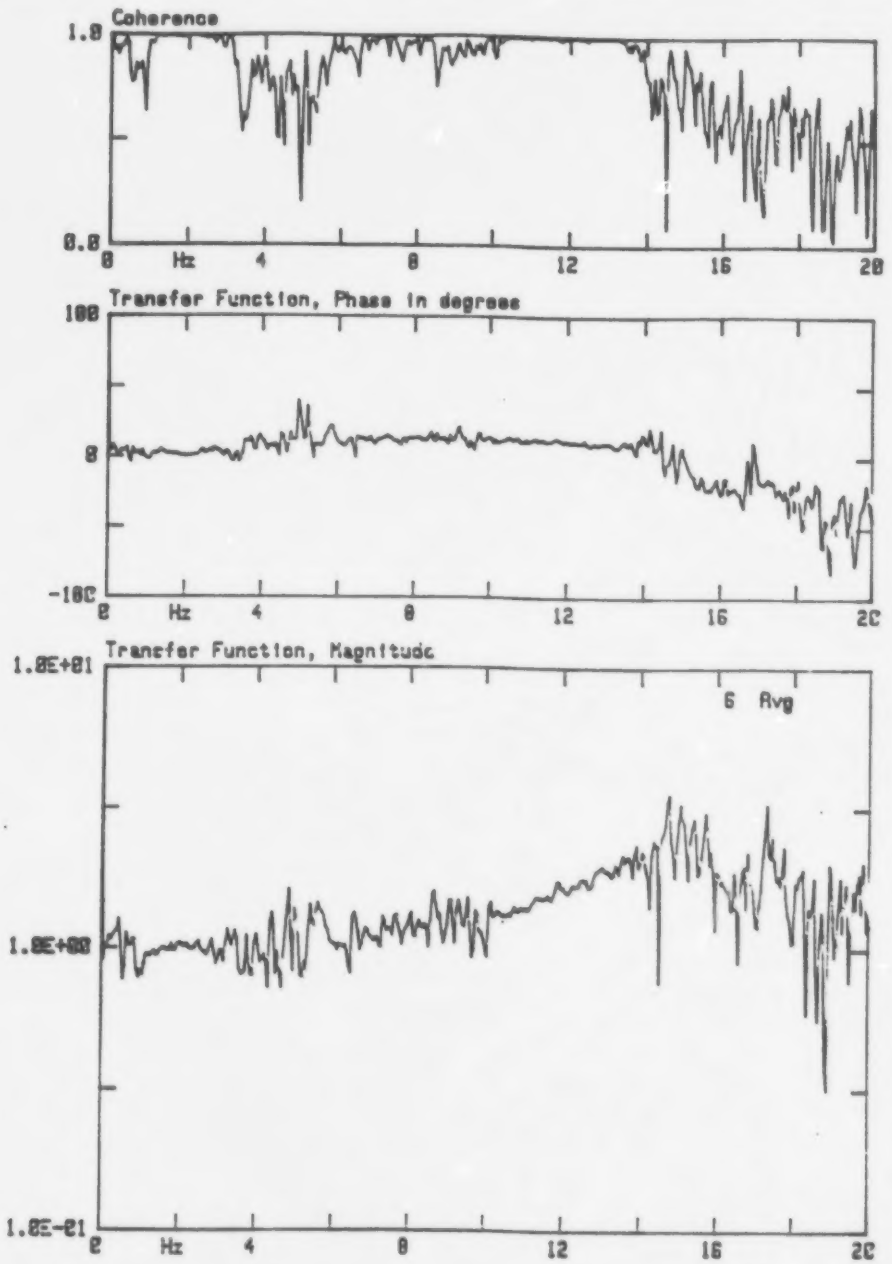
GGC2 , GOK1214_01
Channel A is channel 12, Channel B is channel 14, Port Credit to Clarkson
Start 203:10:13:45, 60C revt out of station, Baseline Data.



GG02 , GOk2318_01
Channel A is channel 23, Channel B is channel 18, Port Credit to Clarkson
Start 283:18:13:45, 600 revs out of station, Baseline Data.



GG02 , GOk1918_01
Channel A is channel 19, Channel B is channel 18, Port Credit to Clarkson
Start 203:18:13:45, 600 revs out of station, Baseline Data.



GOp0008d , GOk_010301, Transfer Function between Upper & Lower Deck
GO Transit Testing 1988.05.21
Channel A is Channel 01, Channel B is Channel 03, Port Credit to Clarkson
10:13:45 to 10:14:45, Baseline Data.

APPENDIX E

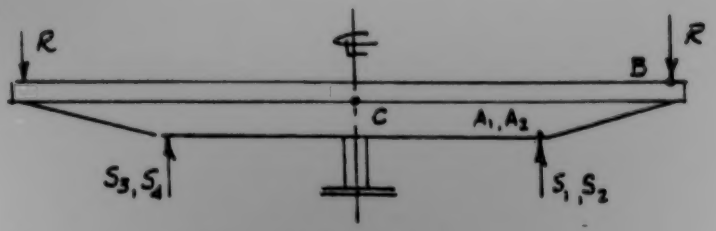
DOFASCO STRUCTURAL DEFLECTION TEST DATA

FOR BOLSTER AND TRUCK FRAME

Bi-Level Ride 88 Oct. 07

Bolster Defection Test

Al G Dofasco



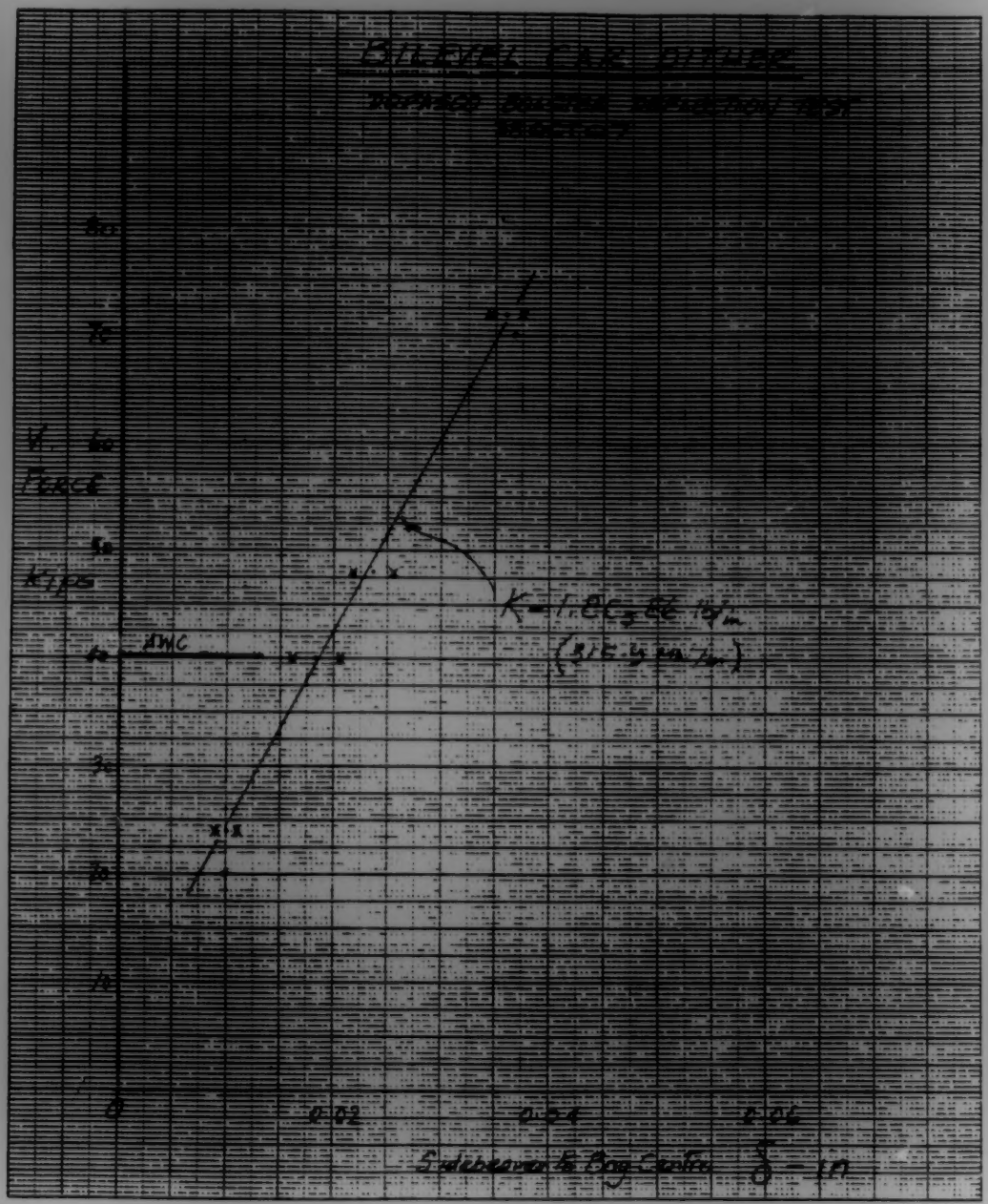
$S_{1 \ldots 4}$ = Sidebearer Force - lb
R = Reaction Force at airbag centres - lb
δ_{A_1B} = Airbag to Sidebearer 1 deflection - in
δ_{A_2B} = Airbag to Sidebearer 2 deflection - in
δ_{BC} = Airbag to Bolster Centre deflection - in

	Force S_1, S_2, S_3, S_4 K-lb	A_1B in	A_2B in	CB in
	0	0	0	0
	6	0.0111	0.009	0.013
AWO:	10	0.0206	0.016	0.0235
	12	0.0255	0.0218	0.0295
	18	0.0375	0.0345	0.0438

At AWO: K_{mean} = 1.805E6 lb/in or 315.9 MN/m

Berence Freq. of Body on Bolster, f_B

$$f_B = \frac{1}{2\pi}\sqrt{\frac{315.9E6}{36615/2}} = 20.9 \text{ Hz}$$


Force
kips
AWC
0.02
0.04
0.06

FILE MEMORANDUM

DATE : December 13, 1988

SUBJECT: Go Bilevel Frame - Vertical Bending Deflection Test

Purpose

To determine the relative bending deflection of the frame due to the application of various vertical loads.

Procedure

The frame was loaded at the sidebearers pads by four interconnected hydraulic jacks (Enerpac RC 302) and reacted at each journal (@ 108" not 102" centers) by four interconnected hydraulic jacks (Enerpac RC 506). The hydraulic pressure was monitored by a Viatran 218-12 pressure transducer.

Four dial gauges were mounted to measure the deflection of the frame at each journal (@ 99" not 102" centers) with respect to a beam supported on the bottom of the frame along the sidebearer centerline as shown on Figure 1 attached.

Results

Load (lb) 'V'	Deflection (x 0.001") Dial Gauge Reading #1	#2	#3	#4
2000	6	5.5	7	7
4000	12.5	12	15	14
6000	17.5	17	20.5	19
8000	23	25	27	26
10000	30	33	34	33
12000	36	40	42	40
14000	42	48	49	46
16000	48	54	55	52
18000	54	61	62.5	59
20000	60	67	65	68

Note: Load is lb/sidebearer

Mark Stewart

MS/ms

cc: J.A. Gaiser
K. Sandhar

File RT 115.7

13 DEC 88

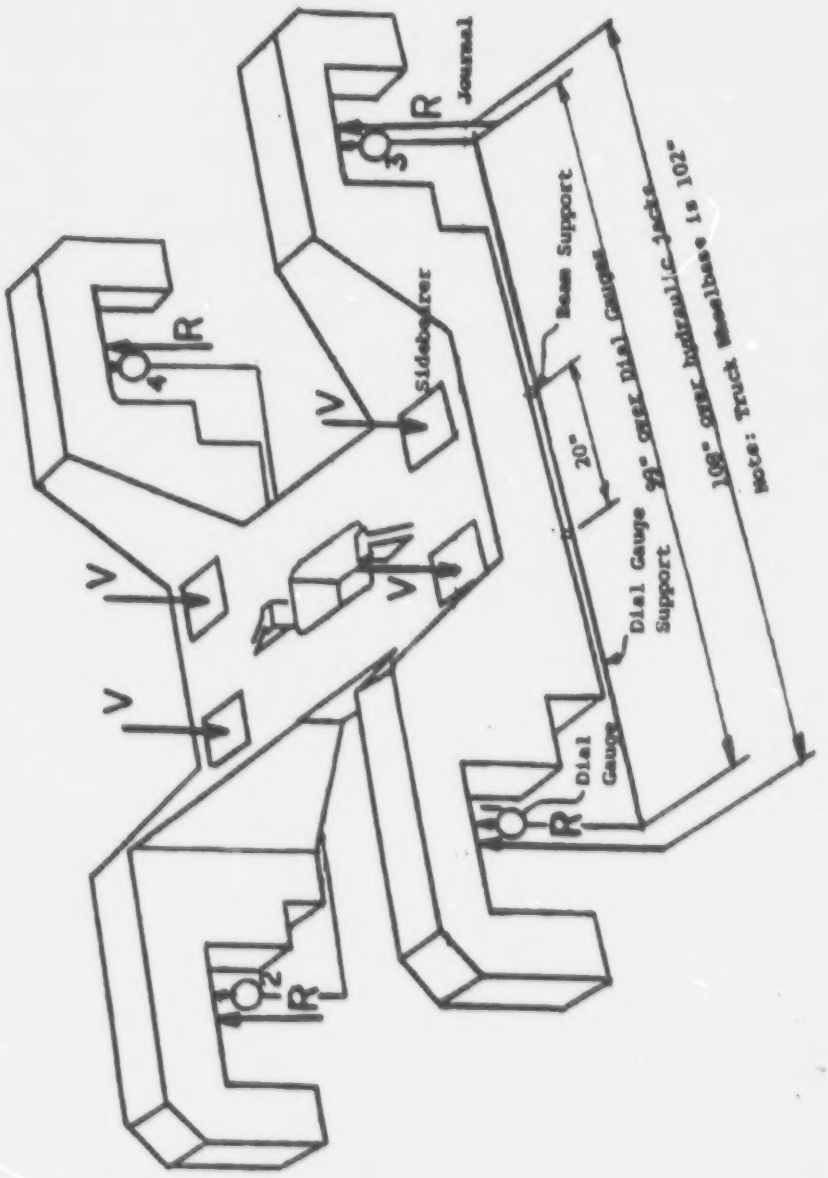


FIGURE 1

Go Bi-level Frame - Vertical Bending Deflection Test

Bi-Level

Improved Definition

Frame Deflection Test
Averaged Readings

ΣV kip	$\Sigma \frac{\delta}{4}$ thou in.
8	6.375
16	13.375
24	18.500
32	25.25
40	32.5
48	39.5
52	46.25
64	52.25
72	59.125
80	65.00

Mean Stiffness about F = 40,000 lb/in

$$K_f = \frac{(48 - 32) \times 10^3}{(39.5 - 25.25)10^{-3}} = 1.123 \times 10^6 \text{ lb/in}$$

$$= 196.5 \text{ MN/m}$$

If $\delta_{tot} = \delta_{bolster} + \delta_{frame}$

$= 0.023 + 0.0325$

$= 0.0555$ in

$$f_{vert} = \frac{3.13}{\sqrt{0.0555}} \sqrt{\frac{40E3}{46.4E3}} = 12.34 \text{ Hz}$$

Using K_f

$$f_{vert} = \frac{1}{2\pi}\sqrt{\frac{K_{tot}}{m}} = \frac{1}{2\pi}\sqrt{\frac{121.1E6}{21,044}} = 12.08 \text{ Hz}$$

$$K_{tot} = \frac{1}{\frac{1}{k_f} + \frac{1}{k_{bolst}}} = \frac{1}{\frac{1}{196.5E6} + \frac{1}{315.9E6}} = 121.1E6 \text{ N/m}$$

APPENDIX F

REFINED BASELINE MODEL

LIST OF MODEL PARAMETERS

A'GEM CHECK FILE - 9:52 AM TUE., 4 JULY, 1989

Input File: BILEVEL.TST
Output File: BILEVEL02.GEM

BILEVEL VERTICAL MODEL #033 - TWO HALF BODIES + BOLSTER/FRAME BENDING MASSES

THERE ARE 27 RIGID BODIES

Body No	C.G. POSITION			MASS	IXX	IXY	IXZ	IYY	IYZ	IZZ	
1	11.8450	0.0000	-.4191	1247.4	576.2	0.0	0.0	56.5	0.0	576.2	Wheelset #1
2	8.4580	0.0000	-.4191	1247.4	576.2	0.0	0.0	56.5	0.0	576.2	Wheelset #2
3	9.7540	0.0000	-.4340	1304.0	531.0	0.0	0.0	1774.0	0.0	3062.0	Fwd Frame
4	9.7540	0.0000	-.5660	499.0	390.0	0.0	0.0	33.9	0.0	618.1	Fwd Bolster
5	5.4780	0.0000	-2.0570	17771.0	39545.0	0.0	0.0	533850.0	0.0	533850.0	Body Half Fwd (Aw0)
6	-8.4580	0.0000	-.4191	1247.4	576.2	0.0	0.0	56.5	0.0	576.2	Wheelset #3
7	-11.8450	0.0000	-.4191	1247.4	576.2	0.0	0.0	56.5	0.0	576.2	Wheelset #4
8	-9.7540	0.0000	-.4340	1304.0	531.0	0.0	0.0	1774.0	0.0	3062.0	Aft Frame
9	-9.7540	0.0000	-.5660	499.0	390.0	0.0	0.0	33.9	0.0	618.1	Aft Bolster
10	11.8450	.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 1-L
11	11.8450	.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 1-R
12	8.4580	-.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 2-L
13	8.4580	.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 2-R
14	-8.4580	-.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 3-L
15	-8.4580	.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 3-R
16	-11.8450	-.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 4-L
17	-11.8450	.7430	0.0000	1.0	0.0	0.0	0.0	0.0	0.0	0.0	Wheel Element 4-R
18	0.0000	0.0000	-2.7000	1873.0	0.0	0.0	0.0	0.0	0.0	0.0	Upper Deck Moving Mass
19	-5.4780	0.0000	-2.0570	17771.0	39545.0	0.0	0.0	533850.0	0.0	533850.0	Body Half Aft (Aw0)
20	10.0840	-1.3210	-.8360	3.2	0.0	0.0	0.0	0.0	0.0	0.0	Secdy Damper Link fwd/lhs
21	9.4240	1.3210	-.8360	3.2	0.0	0.0	0.0	0.0	0.0	0.0	Secdy Damper Link fwd/rhs
22	-9.4240	-1.3210	-.8360	3.2	0.0	0.0	0.0	0.0	0.0	0.0	Secdy Damper Link aft/lhs
23	-10.0840	1.3210	-.8360	3.2	0.0	0.0	0.0	0.0	0.0	0.0	Secdy Damper Link fwd/rhs
24	9.7540	0.0000	-.4720	1304.0	531.0	0.0	0.0	0.0	0.0	0.0	Frame Bending Mass (FWD)
25	9.7540	0.0000	-.6040	499.0	390.0	0.0	0.0	0.0	0.0	0.0	Bolst Bending Mass (FWD)
26	-9.7540	0.0000	-.4720	1304.0	531.0	0.0	0.0	0.0	0.0	0.0	Frame Bending Mass (AFT)
27	-9.7540	0.0000	-.6040	499.0	390.0	0.0	0.0	0.0	0.0	0.0	Bolst Bending Mass (AFT)

THERE ARE 55 DEGREES OF FREEDOM

Body	Lon	Lat	Ver	Rol	Pit	Yaw	
1	0	0	2	1	0	0	Wheelset #1
2	0	0	4	3	0	0	Wheelset #2
3	0	30	6	5	7	0	Fwd Frame
4	0	31	8	9	0	0	Fwd Bolster
5	0	32	10	11	12	0	Body Half Fwd (Aw0)
6	0	0	14	13	0	0	Wheelset #3
7	0	0	16	15	0	0	Wheelset #4
8	0	33	18	17	19	0	Aft Frame
9	0	34	20	21	0	0	Aft Bolster
10	0	0	22	0	0	0	Wheel Element 1-L
11	0	0	23	0	0	0	Wheel Element 1-R
12	0	0	24	0	0	0	Wheel Element 2-L
13	0	0	25	0	0	0	Wheel Element 2-R
14	0	0	26	0	0	0	Wheel Element 3-L
15	0	0	27	0	0	0	Wheel Element 3-R
16	0	0	28	0	0	0	Wheel Element 4-L
17	0	0	29	0	0	0	Wheel Element 4-R
18	0	0	35	0	0	0	Upper Deck Moving Mass
19	0	41	36	42	43	0	Body Half Aft (Aw0)
20	0	0	37	0	0	0	Secdy Damper Link fwd/lhs
21	0	0	38	0	0	0	Secdy Damper Link fwd/rhs
22	0	0	39	0	0	0	Secdy Damper Link aft/lhs
23	0	0	40	0	0	0	Secdy Damper Link fwd/rhs

24	0	44	45	46	0	0	Frame Bending Mass (FWD)
25	0	47	48	49	0	0	Bolst Bending Mass (FWD)
26	0	50	51	52	0	0	Frame Bending Mass (AFT)
27	0	53	54	55	0	0	Bolst Bending Mass (AFT)

THERE ARE 14 POINT CONNECTION ELEMENTS.

Join No	End1	End2	----POSITION----			DIRECTION	STIFFNESS	DAMPING	
1	1	3	11.8498	-.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
2	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
3	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
4	3	"	11.8498	.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
5	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
6	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
7	2	3	8.4588	-.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
8	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
9	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
10	2	3	8.4588	.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
11	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
12	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
13	6	8	-8.4588	-.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
14	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
15	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
16	8	"	-8.4588	.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
17	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
18	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
19	7	8	-11.8498	-.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
20	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
21	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
22	7	8	-11.8498	.5988	-.4198	Lon	8.9758E+87	2.4188E+84	Chevron Primaries (5.8% Crit
23	"	"	"	"	"	Lat	1.3418E+87	9.3458E+83	
24	"	"	"	"	"	Ver	5.5688E+86	7.8888E+83	
25	4	4	9.7548	-.5598	-.4888	Lon	1.8288E+85	7.1388E+82	Sidebearer Pads (5.8% Crit D
26	"	"	"	"	"	Lat	2.5288E+85	1.1218E+83	
27	"	"	"	"	"	Ver	1.4888E+87	8.3588E+83	
28	4	4	9.7548	.5598	-.4888	Lon	1.8288E+85	7.1388E+82	Sidebearer Pads (5.8% Crit D
29	"	"	"	"	"	Lat	2.5288E+85	1.1218E+83	
30	"	"	"	"	"	Ver	1.4888E+87	8.3588E+83	
31	9	9	-9.7548	-.5598	-.4888	Lon	1.8288E+85	7.1388E+82	Sidebearer Pads (5.8% Crit D
32	"	"	"	"	"	Lat	2.5288E+85	1.1218E+83	
33	"	"	"	"	"	Ver	1.4888E+87	8.3588E+83	
34	9	9	-9.7548	.5598	-.4888	Lon	1.8288E+85	7.1388E+82	Sidebearer Pads (5.8% Crit D
35	"	"	"	"	"	Lat	2.5288E+85	1.1218E+83	
36	"	"	"	"	"	Ver	1.4888E+87	8.3588E+83	
37	4	4	9.7548	8.8888	-.4278	Lon	1.1558E+87	6.2588E+83	Traction Pads
38	"	"	"	"	"	Lat	6.2588E+84	1.1888E+81	
39	9	9	-9.7548	8.8888	-.4278	Lon	1.1558E+87	6.2588E+83	Traction Pads
40	"	"	"	"	"	Lat	6.2588E+84	1.1888E+81	
41	25	5	9.7548	-1.1188	-.8518	Ver	6.6398E+85	8.8888E+88	Airsprings
42	25	5	9.7548	-1.1188	-.8518	Ver	6.6398E+85	8.8888E+88	Airsprings
43	27	19	-9.7648	-1.1188	-.8518	Ver	6.6398E+85	8.8888E+88	Airsprings
44	27	19	-9.7548	-1.1188	-.8518	Ver	6.6398E+85	8.8888E+88	Airsprings
45	28	25	10.8848	-1.3218	-.8368	Ver	3.6888E+87	8.8888E+88	2ndary Vert Damper Link Bush
46	21	25	9.4248	-1.3218	-.8368	Ver	3.6888E+87	8.8888E+88	2ndary Vert Damper Link Bush
47	22	27	-9.4248	-1.3218	-.8368	Ver	3.6888E+87	8.8888E+88	2ndary Vert Damper Link Bush
48	23	27	-10.8848	-1.3218	-.8368	Ver	3.6888E+87	8.8888E+88	2ndary Vert Damper Link Bush
49	28	5	10.8848	-1.3218	-.8368	Ver	8.8888E+88	5.5348E+84	2ndary Vert Dampers (36.X C
50	21	5	9.4248	-1.3218	-.8368	Ver	8.8888E+88	5.5348E+84	2ndary Vert Dampers (36.X C
51	22	19	-9.4248	-1.3218	-.8368	Ver	8.8888E+88	5.5348E+84	2ndary Vert Dampers (36.X C
52	23	19	-10.8848	1.3218	-.8368	Ver	8.8888E+88	5.5348E+84	2ndary Vert Dampers (36.X C
53	24	4	9.3868	8.8888	-.6968	Lat	8.8888E+88	4.4188E+84	2ndary Lat Dampers
54	26	9	-9.3868	8.8888	-.6968	Lat	8.8888E+88	4.4188E+84	2ndary Lat Dampers
55	1	10	11.8498	-.7438	8.8888	Ver	2.8888E+87	5.3888E+83	Wheel Element Springs

| | | |

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Volume II - Test Report

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Prepared by:
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March 1990

Technical Report Documentation Page

Bi-level III Ride Quality Improvement Program
Final Report – Volume II - Test Report

Prepared by:	**UTDC Inc.**
Prepared for:	Rail and Advanced Transportation Systems Office Transportation Technology and Energy Branch Ministry of Transportation T. AppaRao, Project Manager
Report No.:	RATSO-90-01
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Abstract:	The objective of the program was to improve the vertical ride quality of the GO Transit Bi-level passenger car upper deck. Three proposed modifications to the car were designed and fabricated: a) stanchions connecting the upper and lower decks, b) vibration absorber, and c) secondary damper link bush modification. Initial tests were carried out to measure the ride quality of a sample of cars in the fleet to establish a baseline. The modifications were installed in a car and were initially tested statically with an impact hammer, then dynamically over characteristic track sections. Transducers (both vertical and lateral) were used to measure accelerations on both the upper and lower deck.
Key words:	commuter rail, passenger car, ride quality, testing and ride measurement
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EXECUTIVE SUMMARY

The objective of this program was to improve the ride quality of the GO Transit Bi-level passenger car upper deck.

The upper deck was found to be subject to vertical vibration (12-16 Hz) known as "dither" during operation at high speeds. The major constraints imposed on the program were a) to improve the vertical ride quality of the upper deck such that it met a mutually acceptable standard (i.e., 2.5 hours I.S.O. reduced comfort boundary), and b) any modification devised had to be less than $10K on a production basis.

Initial ride quality tests were carried out on a sample of cars to establish a baseline. Three proposed modifications to the car design were then designed and fabricated, namely a) stanchions connecting the upper and lower decks, b) a vibration absorber, and c) a secondary damper link bushing modification. The above modifications were installed in a car and were initially tested with an impact hammer. The modifications were then tested dynamically in revenue service over characteristic track sections. Vertical and horinzontal transducers were used to measure accelerations on both the upper and lower deck of the test car.

The stanchions reduced the vibration in the 16 Hz mode, but showed little improvement for the 12 Hz bending mode. The vibration absorber reduced the dither vibration by 35%. The secondary damper link bushing modification yielded a very small improvement.

It was deemed that none of the modifications yielded ride quality improvements significant enough to warrant a retrofit of the fleet.

TITLE: BI LEVEL III RIDE QUALITY IMPROVEMENT PROGRAM FINAL REPORT VOLUME II TEST REPORT

1.0 INTRODUCTION

1.1 BACKGROUND

The Bi-level III Ride Improvement Program was initiated in May 1987 after completion of negotiations between MTC/UTDC and GO Transit. It had as its purpose the elimination of the unpleasant upper deck vertical vibration known as "dither" which exists on the GO Bi-level commuter rail car. This vibration problem has been investigated at various times by teams from Hawker Siddeley (now Can Car Works, Thunder Bay), Dofasco and MTO. In 1987 March, discussions amongst MTO, GO and UTDC resulted in a two-phase program being agreed to which would address the dither problem. Phase I of this program would see two proposed fixes: a) replacement of the upper deck floor with strengthened panels and b) application of vertical primary dampers between the axle boxes and truck frame. If no acceptable solution was found in Phase I, then a more in depth analysis effort would be considered in a Phase II which would be negotiated separately. Each proposed modification to the car was to be evaluated analytically and then in a static shop test. If judged suitable, then the mod would be road tested. The chief constraint on the program was that the car modification cost on a production basis had to lie within 10K$. The program was completed in February 1989 and consisted of analytical as well as test portions. This report describes the test activities which were carried out.

The data available at the start of the program suggested that dither was centred on the upper deck and occurred over a frequency range from 10 - 15 Hz. There was also evidence that the truck frame primary bounce frequency and wheelset rotational frequencies were in the same range and thus could excite the dither vibration. These early tests investigated the effect of wheel unbalance, drag link position, airbag stiffness and damping and shock absorber configuration. In the case of the latter, it was discovered that the dither vibration could be reduced by removing the shock absorbers or reducing shock absorber mounting bushing stiffness. However, concerns over effects upon carbody roll displacement and fatigue life of shock absorber bushings quieted any further work.

1.2 OBJECTIVES

The objectives of the program were:

a) to improve the vertical ride of the car, particularly the upper deck such that it met a mutually acceptable standard (say, 2.5 hours I.S.O. reduced comfort boundary); and

b) to ensure that the cost in production of fitting any modification package to a revenue car would be less than 10 K$;

c) to improve the roll characteristic of the car when negotiating turn-outs and cross-overs (to be done once dither problem resolved).

1.3 METHOD OF TESTING

Since we are dealing with a vibration problem which affects a particular frequency band with a particular amplitude, there were two approaches to measuring its condition when modifications were made. The first was to measure the amplitude over the frequency range of interest and the second was to measure the frequency shift occurring in the two modes of excitation (body vertical bending and upper deck heave). The former approach required accelerometer instrumentation and a capability of converting time domain signals to the frequency domain. Both could be carried out in a static test using the impact hammer technique [1] to gather force and velocity data from which mechanical impedance (force divided by velocity) could be obtained. The troughs of impedance in the frequency domain identify resonance areas, therefore any improvement in dither vibration would be indicated by a frequency shift upwards of one of the troughs or an increase in impedance level.

The test plan called for an initial shop evaluation to be made of each proposed modification using the impact hammer technique. If the results of this test were promising, then a dynamic road test would be carried out in which the ride quality of the three decks would be evaluated with lateral and vertical accelerometers. The results of this test would be compared with the unmodified car (same as test car) to determine the improvement achieved.

2.0 TEST PLAN

The test program was organized into three groups. The first two, initial ride tests of a sample of Bi-level cars in service, and, modification tests were planned for in the original proposal [2]. The third group, Improved Definition of Problem test, was added later. Within the second group tests of modifications, an increase in the number of mods examined occurred part way through the program [3].

2.1 INITIAL RIDE MEASUREMENTS

Before starting work on the program, baseline data had to be gathered on several Bi-level coaches under service conditions. Termed the 'travelling road show', this task was carried out by MTO in 1987 April. A total of 18 coaches (including 3 cab cars) were evaluated for ride at the centres of the upper deck and lower deck. For this, MTO provided a portable data acquisition system consisting of two sets of vertical and lateral accelerometers positioned on the upper and lower decks, feeding an FM multi-plexer and tape recorder. Recordings were made of the unweighted lateral and vertical acceleration on each deck over the roughest track zone between Port Credit and Oakville Stations at normal operating speeds. From this data, a worst case car would be selected and used as the test car in the modification evaluation work.

2.2 TESTS OF MODIFICATIONS

The plan in the proposal for modification tests included only two items 1) a maximum of two stiffened floor panel designs and 2) truck primary suspension dampers. After encouraging test results were obtained with stanchions on an unfinished carshell, it was decided to include stanchions in the list of mods to consider and to give priority to a stanchion road test. The stanchions were tested in 1987 December and again in 1988 February in a subjective juried test [5] and because of their disappointing performance at improving car ride it was obvious that other further changes were needed to be made to the test program. At this point 1988 March 01, it was decided to obtain more data on the problem and the 'Improved Definition of Problem' test was conceived. This test would take a closer look at truck effects upon dither vibration since it appeared that not only was the body tuned to the two key frequencies of 12 Hz and 15 Hz, but the truck appeared capable of exciting these as well. Two other mods not previously considered were introduced at this time for assessment. They were the vibration absorber

and the modified secondary vertical damper bush for the Houdaille units. The latter two were lumped together in the improved definition test to facilitate test planning. The tests on these three items were carried out by 1988 June.

2.3 IMPROVED DEFINITION OF PROBLEM TEST

The improved definition of problem test was to be conducted in three parts:

a) baseline car test
b) vibration absorber test; and
c) secondary damper link bush mod test.

The Baseline car test was organized to provide a two day period to install truck and body instrumentation followed by shop verification tests, then a road test to clear any snags in the system prior to actual data collection. The baseline tests were planned over a two day period and were to cover the three test zones:

1. Scarborough - Danforth Stations
2. Pt. Credit - Clarkson Stations
3. Clarkson - Oakville Stations.

During the runs, areas of 'good' track where no significant dither response occurs, were also to be recorded. Vehicle speeds were to be taken as operational levels occuring during the runs. A quick look capability was to be provided by MTO to facilitate assessment of data quality and performance trends during testing.

The vibration absorber was to be tested on Car 2030 by attaching it to the floor brackets of the four central stanchions on the lower deck.

A static calibration was first to be done to correlate the absorber mass level against the beam strain gauge output by adding weights one at a time (mean weight 66.8 kg). The plan then called for recording the car response, over the three test zones, from three different weight settings (8, 10 and 11 weights).

Finally a combined stanchion/vibration absorber test would be run with the 8 stanchion arrangement with both 10 weights and 11 weights on the absorber.

The secondary shock absorber mod development was originally planned to be carried out on a Houdaille equipped car. This would have necessitated a detailed bench test at MTO to finalize the bushing configuration. Because of difficulty in mounting Houdaille to Koni equipped test car and unavailability of resources to carry out this phase of the bushing modification program, it was decided to use the Koni shock absorber installation on Car 2030 and to try to duplicate the original HSC test using the recommended configuration [4]. However, prior to this decision to abandon the Houdaille mod, a one day ride comparison test was made of seven Bi-level Mk II cars to select a suitable test car for the Houdaille bushing mod evaluation.

The test plan then called for a run with the modified bush arrangement over the three test zones followed by combined runs in which bushing mod plus vibration absorber plus stanchions mod were to be evaluated.

3.0 RIDE CRITERIA

At the beginning of the program, it was mutually agreed amongst GO, MTO and UTDC that a set of target ride performance specifications be derived and used to guide the evaluation of proposed car mods. The result of discussions amongst the three groups is summarized below.

3.1 PURPOSE

The following Target Performance Specification was developed with a view to providing:

a) a quantitative measure to evaluate the success of the project,
b) a guideline for designing floor, suspension or other modifications, and
c) a quantitative definition of the scope of the project.

3.2 TARGET PERFORMANCE SPECIFICATIONS

The vertical ride quality at any location on either the upper or lower deck of the bi-level car shall be no less than 2.5 hours ISO when measured and evaluated as follows:

a) vehicle speed for testing shall be 80 mph;

b) measurements shall be taken with the Test Vehicle travelling on the Port Credit - Oakville - Port Credit portion of the GO Transit Lakeshore Line, Canadian National Railway's Oakville Subdivision Milepost 12.8 to Milepost 22.4;

c) the ISO ride quality shall be measured and evaluated as per ISO Standard #2631;

d) the worst one-minute segment of the test run will be used for evaluation purposes; and

e) the AW0 load condition is used.

4.0 INSTRUMENTATION ARRANGEMENT

The following section describes the instrumentation arrangement used during: a) the initial ride survey, b) modification test of stanchions, and c) improved definition of problem test.

4.1 INITIAL RIDE SURVEY

The initial ride survey, conducted on 18 bi-level cars, used a simple arrangement of accelerometers as follows:

Channel No.	Transducer	Direction	Located
1	Accelerometer	Vertical	Upper deck car centre near seat support.
2	Accelerometer	Lateral	Upper deck car centre near seat support.
3	Accelerometer	Vertical	Lower deck car centre near seat support.
4	Accelerometer	Lateral	Lower deck car centre near seat support.

Data was recorded on tape and processed later at MTO's laboratory.

4.2 MODIFICATION TESTS OF STANCHIONS

Because the stanchion mod was tested separately from the rest of the mods a different instrumentation arrangement was used. Two distinct test arrangements were used here: a) one for the impact test in the shop and b) the other for the dynamic road tests.

4.2.1 Impact Test

The impact test instrumentation is the same as used in [1] and consists of:

1. Loadcell, Kistler model 912
2. Accelerometer, Bruel and kjaer model 4370
3. Two Charge Amplifiers, Bruel and Kjaer model 2635
4. A Tunable Filter, Krohn-Hite model 3343
5. A Dual Channel FFT Analyser, Nicolet model 660A
6. A Data Recorder, Nicolet model 1608.

The accelerometer locations and the point of input are governed by the actual structure of the railway coach and are shown in Figure 1.

The hammer is used to excite the structure of the coach at each set of input and output locations. Both, the input to, and the output from the structure are measured by the FFT analyser. The structure is excited at least five times at each set of measuring points, to verify the reproducibility of the experiment. The data for each set of points is stored on disk after it has been judged to be acceptable.

4.2.2 Road Test

The stanchion road test was carried out with car 2030 using the following instrumentation.

Channel No.	Description
1	Vertical Acceleration Upper Deck centre
2	Vertical Acceleration Lower Deck centre
3	Vertical Acceleration Intermediate Deck centre
4	Lateral Acceleration Upper Deck centre
5	Lateral Acceleration Lower Deck centre
6	Lateral Acceleration Intermediate centre
7	Strain of Stanchion 1
8	Strain of Stanchion 2
9	Strain of Stanchion 3
10	Strain of Stanchion 4
11	Strain of Stanchion 5
12	Strain of Stanchion 6
13	Vertical Acceleration of forward bolster - r.h.s.

Further details are available from MTO on specific recording equipment used.

4.2.3 Juried Road Test

After the initial instrumented road test it was decided to carry out a subjective ride test with a selected jury of five people [5]. No instrumentation was used except for a prepared form in Figure 2 given to each jury member. The approach used was to ride a reference unmodified Car (2001) and then step into Car 2030 with the stanchions installed and record ride impressions. Data on the forms was then later processed to determine the perceived improvement from the stanchions.

4.3 IMPROVED DEFINITION TEST

Since we were interested in a wider range of data, the improved definition test was the most elaborate one of three tests done.

Channel Configuration

Channel 1	Vertical acceleration on centre of the lower deck.
Channel 2	Lateral acceleration on centre of the lower deck
Channel 3	Vertical acceleration on centre of the upper deck
Channel 4	Lateral acceleration on centre of the upper deck
Channel 5	Vertical acceleration on centre of the interm. deck
Channel 6	Lateral acceleration on centre of the interm. deck
Channel 7	Vertical acceleration of the left axle box
Channel 8	Vertical acceleration of the right axle box
Channel 9	Vertical acceleration of the r.h.s. of interm. deck
Channel 10	Longitudinal acceleration of left bolster
Channel 11	N/A
Channel 12	Vertical acceleration of right frame, forward
Channel 13	N/A
Channel 14	Vertical acceleration of left frame, aft
Channel 15	Vertical acceleration of left side of bolster
Channel 16	Pressure in airbag (left)
Channel 17	N/A
Channel 18	Longitudinal acceleration of right side of bolster
Channel 19	Vertical acceleration of right side of bolster
Channel 20	Strain gauge on vibration absorber beam centre
Channel 21	N/A
Channel 22	Longitudinal acceleration of left frame, aft
Channel 23	Vertical acceleration of right frame, aft
Channel 24	Longitudinal acceleration of right frame, aft.

All sensitivities etc. at 50 Hz.

5.0 DATA REDUCTION METHODS

This section describes the methodology of reducing the raw data to Engineering meaningful units for each of the major tests.

5.1 INITIAL RIDE MEASUREMENTS

The vertical raw accelerometer output from the initial ride test of eighteen cars was processed for both the upper and lower decks. The data was passed through a two channel analyser to produce 10 minute and 1 minute averaged spectra. Only the data from the zone from Port Credit to Oakville zone was analysed. The data was then presented as a plot of r.m.s. acceleration in m/sec2 versus frequency up to 20 Hz and compared with standard I.S.O. exposure limits of 1,2.5,4, 8,16 and 24 hours. A summary plot comparing the weighted overall r.m.s. acceleration of each car was created for both 10 minute and one minute averaging periods. In addition to this, plots of the transfer function and coherence between lower and upper deck acceleration were also made (See Appendix B, Volume I).

5.2 ROAD TEST FOR STANCHION MODIFICATION

The data from the stanchion tests was reduced according to the type of test done: a) Impact hammer test and b) road test.

5.2.1 Impact Hammer Test

The data from the impact test was collected in sets consisting of input force and simultaneous acceleration response. The position of the response point did not change during the test while the impact position varied according to Figure 1. The basic approach was to create the mechanical impedance function by dividing the force spectrum by the singly integrated acceleration spectrum for a particular test position giving, in effect, force divided by velocity. A resonant point therefore would show up as a dip in the impedance curve and a high impedance was a sign of good behaviour. The purpose of this test was to compare the impedances of various stanchion arrangements with a view to picking the best one. The impedance of every impact location was derived from an average of at least five measurements. Each location was examined in assessing the effectiveness of a particular stanchion arrangement.

5.2.2 Road Test

The stanchions were tested in a road test in 1987 December and from this data a final technical assessment was made. To the instrumentation arrangement used in the initial ride tests was added the intermediate deck lateral and vertical acceleration above the forward truck and axial strain gauge measurements on six stanchions. The data from this test was reduced similarly to that in the initial road tests in terms of I.S.O. plots of unweighted acceleration versus frequency.

5.3 IMPROVED DEFINITION TEST

A vast amount of data was collected on tape in the 1988 June improved definition test. To obtain a first look and basic understanding of the car behaviour a simplified approach was needed. As a result it was decided to concentrate on the zone between Port Credit and Clarkson (Eastbound and Westbound directions) and to look at the narrow band spectra and transfer functions between selected pairs of channels. By checking the coherence of any selected channel pairs, one could decide if there was a linear cause and effect relationship between them (high coherence). If no such effect existed (low coherence) then either: a) a non-linear relation existed between them (difficult to identify) or b) self generated motion was occurring at one of the channel locations. Use of phase angle information in conjunction with transfer function magnitude was used to identify mode shapes where possible. The possibility of doing a conventional modal analysis on the entire car was ruled out because: a) the track input was not truly constant (stationary and ergodic) and b) no convenient source of large force actuators was available with which to do a shop test. By examining a number of transfer function plots between different acceleration responses etc. measured from the axle up to the top deck, it was possible to get an impression of how the truck and carbody behaved. Most of this data can be seen in Appendix D of Volume 1. Other data not analysed is in storage at MTO under the care of J. Tiessinga.

6.0 TEST RESULTS

The results from the shop and road tests conducted as part of this program are briefly discussed. Reference to Volume I of this report will be made at intervals to avoid duplication of data in this part of the report.

6.1 INITIAL RIDE QUALITY

The initial ride survey was done in 1987 April to get an impression of the average state of the fleet plus an idea of which car, a bad actor, would be suitable for use as a test car in the nest of the program. Examination of the results in Appendix B, Volume I indicate that the worst performer was car 2030, a Mk I car equipped with Koin shock absorbers. In Figure 3, the fleet generally shows better than a 4 hour ride on the lower deck but only 1 to 2.5 hours on the upper deck. The latter is primarily due to a peak associated with dither vibration centred on the 12.5 Hz third octave band.

6.2 STANCHION MODIFICATION

The stanchion test results are organized into two sections: a) shop test and b) road test.

6.2.1 Shop Tests

The stanchions were first investigated in a shop test [6] on a bare carshell at CCW in 1987 July. In this test, two sets of jack poles were placed between upper and lower deck, and, upper deck and roof cross-beams in a 4 square arrangement. The floor structure was excited by dropping a rubber tire and the resulting vertical acceleration response was recorded. Seven test configurations were measured to compare upper floor and adjacent window sill responses in the as - is and modified conditions. When data was expressed in the frequency domain, three sets of peaks occurred, 11 Hz, 17 Hz and 22 - 30 Hz; referred to as body bending, dither and super dither, respectively. Comparison of data showed, when normalized against the body bending response in each case (see Figure 4), that a significant (up to 30:1) reduction of upper deck vibration amplitude at dither frequency could be achieved. A 15:1 reduction was found for the super dither mode.

This test provided the motivation to look more thoroughly at the stanchion mod.

The second series of shop tests were done in 1987 December on test car 2030 at Willowbrook Shop. This time a more practical stanchion installation (UTDCI Drawing No. EX215-664-001) was incorporated using stainless steel pipe and standard floor and ceiling brackets.

the purpose of this test was to determine the overall effect of stanchions upon upper and lower deck natural frequencies and modes and to select an optimum arrangement of stanchions which would be evaluated in a future road test.

MTO was called upon to perform the test using the impact hammer technique (1). To do this, 11 stations were selected on the upper deck and, 6, on the lower deck as shown in Figure 1. The vertical impedance function was calculated for each point for each of the following stanchion configurations:

Configuration #	Description
1	Unmodified Car
2	4 Square Stanchion Arrangement
3	4 Zig-Zag Stanchion Arrangement
4	6 Staggered Stanchion Arrangement

The resulting impedance plots are shown in Figures 5 through 10. Curves are numbered to indicate which configuration from the list above applies. Coherence blanking to a level of 0.9 has been used in obtaining each plot. This removes any data which is lying at a coherence level less than 0.9 and ensures that only meaningful data is retained. The location on the car from Figure 1 is given by the last three digits of the file name enclosed by each plot in Figures 5 through 10.

Generally, the upper deck is affected more than the lower deck when stanchions are installed and the frequency band affected is rather broad, from 10 - 40 Hz. The six staggered stanchion arrangement is most effective, closely followed by the four zig-zag arrangement. The benefits, occur at 16 Hz and range from 2.8:1 to 4.8:1 depending upon location, and, reduce as one moves away from the centre of the deck. The natural frequencies tend to be lower from this test compared to the previous CCW test owing to the fact

that the latter test was done on a bare carshell. The geographic variation of impedance in the latest shop test is similar to that in the test of car 2142 in 1986 January (1). Thus there were sufficient grounds at this stage to perform a road test on the stanchion mod equipped test car.

6.2.2 Road Test

The stanchion mod was evaluated in a road test on th GO Lakeshore line in 1987 December 16 and 17. Out of 13 data channels, five failed to record, however, all the vertical acceleration channels were successfully recorded.

Three stanchion arrangements were evaluated in the east and westbound directions between Port Credit and Oakville, one of three previously selected test zones. The stanchion configurations tested included:

1. No stanchions installed (OUT)
2. Six staggered stanchions arrangement (IN)
3. Four Zig-Zag stanchions arrangement (ALT)

Subjectively, the stanchions made a difference to the upper deck vibration levels, but there was still enough remaining excitation which was noticable enough on rougher track to be called unpleasant. Good track did not excite dither strongly. The vertical acceleration data from upper and lower decks was reduced to narrow band spectra, ride spectra (ISO format) and transfer function between lower and upper decks.

Figure 11 is a summary of the ride results from the road test. With stanchions installed (either configuration) both decks have essentially equal ride on both decks. The baseline car (EB OUT and WB OUT) had a better ride than the same car seven months earlier in the initial ride test (compare "x" with "o") indicating different speed, car, loading and/or roadbed conditions. The four stanchion zig-zag arrangement, unexpectedly gave a better performance than the six stanchion arrangement. This is likely because of speed differences. Certainly, the number of passengers (0) on the four stanchion test should have made the dither worse than the six stanchion configuration. The data does show a significant amount of scatter for the amount of testing done and there is no clear

advantage from stanchions which emerges. This is because improvement is limited to frequencies above 14 Hz as seen in Figure 12. Figure 12 shows the transfer function of upper deck vertical acceleration divided by lower deck acceleration from the initial ride test in 1987 April.

Figure 12b shows the baseline test with no stanchions in 1987 December and 12 c is the test of the six stanchion arrangement in 1987 December. It can be appreciated that if the first body bending mode at 12 Hz affects both decks equally, then the transfer function will stay close to unity with or without stanchions. This is, in fact, the case and underlines the basic failing of stanchions. They do not and can not reduce the first bending mode, the major source of the dither problem.

A check of stanchion stresses during testing indicated peak values below 50 $\mu\varepsilon$ or 10 MPa (1.5 ksi) which is well below the endurance limit.

Stanchions do help but the degree of improvement is not large enough to warrant fleet retrofit.

6.2.3 Juried Test

To settle the question of stanchion effectiveness from the passenger's viewpoint, it was decided to run a completely subjective ride test in 1988 February. The form in Figure 2 was used by a five-man jury (N. Kuster/(GO), R. Lawrence/(GO), R. Johns/(GO), R. MacKenzie/(GO), J. Young/(MTO)) to record their impressions of the difference between a control car (2001) and the stanchion equipped car (2030). Two runs through each of the three standard test zones were made in each direction for a total of 12 runs. The results are shown in Tables 6.2-1, 6.2-2 and 6.2-3. The key result is that the net improvement per direction per zone is only 0.91 on a scale of 0 to 5 while the standard deviation per direction per zone is 1.69. This means that no clear agreement on the benefit of the stanchion mod was reached by the jury - a not too surprising result in view of the technical evaluation in 6.2.2.

SUMMARY OF RESULTS

JURY MEMBER NO.	NO STANCHIONS					STANCHIONS 'IN'				
	WB			EB		WB			EB	
	ZONES 1	2	3	1	2	1	2	3	1	2	3
TEST/ZONE	1/1	1/2	1/3	2/1	2/2	2/1	2/2	2/3	1/1	1/2	1/3
A	0	2	0	-1.5	2.5	-2.5	4.5	-1.5	0	1	-2
B	2	0	1.5	0	3.5	2	5.5	4	3	1.5	3
C	0	1.5	0.5	-2	3	2	3	-1	1.5	0	2.5
D*	0.5	1	-	-	-	-	-	4	-	-	-
E	0.5	2	0	0	0.5	2	2.5	0.5	0	0	0
TOTALS	3	6.5	2	-3.5	9.5	3.5	15.5	6	4.5	2.5	3.5
JURY MEAN	0.6	1.3	0.5	-0.875	2.38	0.875	3.88	1.2	1.13	1.63	0.88
STANCHIONS IN-OUT	-	-	-	-	-	0.25	2.50	0.70	2.00	-1.76	1.68
No. Pass (2030/ 2001)	24/10	20/8	7/7	15/31	14/22	11/6	5/5	7/3	17/11	10/16	8/11	24/18

READINGS ARE IN 1/8" UNITS FROM '0' ON SCALE FROM 0 TO 5 FOR CAR 2030 REL 2001.

(+) = BETTER, (-) = WORSE

*THESE RESULTS HAVE BEEN AVERAGED.

TABLE 6.2-1

TABLE 6.2-2

NET IMPROVEMENT

	TOTAL SUBJECTIVE RATING
SUM STANCHIONS IN SUM STANCHIONS OUT	8.58 3.10
TOTAL NET IMPROVEMENT	5.48
NET IMPROVEMENT/DIR'N/ZONE	0.91

TABLE 6.2-3

RANKING OF IMPROVEMENT

ZONE	DIR'N	TOTAL SUBJECTIVE RATING
2	WB	2.58
1	EB	2.00
3	EB	1.68
3	WB	0.70
1	WB	0.28
2	EB	-1.76
TOTAL		5.48
STD DEVIATION		1.69

6.3 IMPROVED DEFINITION TEST

The improved definition of problem test, run in 1988 June, consisted of three components; a) baseline test, b) vibration absorber test and c) Koni secondary damper bush test. This section discusses the baseline test.

The original intent of this test was to define the energy flow from the wheelsets up to upper deck. This would imply using force and motion transducers at major interfaces. An examination of a truck at Willowbrook in 1988 April 19 suggested that force transducers would be practical for the airbag (air pressure transducer) and secondary damper (Houdaille link could be strain gauged; Koni not so straightforward). In the end, only the pressure transducer was installed as the car was available for a relatively short time and no truck disassembly was possible. Thus the data would consist mainly of motion (acceleration) measurements. The test progressed as shown in Table 6.3-1 below.

TABLE 6.3-1

Date	Activity	Problems
88 June 13, 15	MTO installed instrumentation in Car 2030 at Willowbrook Yard. All 21 channels checked.	Wheel Rev counter not hooked up and velocity not working.
88 June 16	Shakedown Runs. Installed frame for vibration absorber.	15V power supply faulty 2 Axle accelerometer Pre-amps failed.
88 June 17	New Pre-amps installed. Wheel RPM transducer installed Shakedown run.	
88 June 20	Shakedown Run Pressure transducer fixed and low pass (300 Hz) filter added to axle accelerometers. Recorded Baseline Data: Danforth to Rouge Hill Rouge Hill to Danforth Port Credit to Oakville Oakville to Port Credit Port Credit to Willowbrook	Pressure transducer not working; axle accelerometers using too much dynamic range of amplifiers.

TABLE 6.3-1 (Cont'd)

88 June 21	Recorded baseline data: Exhibition to Oakville Oakville to Port Credit	LHS axle accelerometer is intermittant.

The data collected in the baseline runs was stored on mag tape and is stored at the MTO, Transportation and Technology Branch, laboratory. This data was used to refine the basic car vertical dynamic model (Bilevel 8, 9 in Volume I) and to provide a baseline with which to compare the vibration absorber and damper bush mods. The data was processed and output in the form of narrow band spectra (see Appendix D, Volume I) and ride spectra. The latter are presented in this section in Figures 21 through 32.

These are organized as follows:

Fig. No.	Direction	Zone	Narrow Band ChA/ChB	Transfer Fcn ChA/ChB	ISO Plot ChA/ChB
21	Westbound	Port Credit - Clarkson	1/3		
22	Westbound	Port Credit - Clarkson	3/5		
23	Eastbound	Clarkson - Port Credit	1/3		
24	Eastbound	Clarkson - Port Credit	3/5		
25	Westbound	Clarkson - Port Credit		1/3	
26	Westbound	Clarkson - Port Credit		3/5	
27	Eastbound	Clarkson - Port Credit		1/3	
28	Eastbound	Clarkson - Port Credit		3/5	
29	Westbound	Clarkson - Port Credit			1/3
30	Westbound	Clarkson - Port Credit			3/5
31	Eastbound	Clarkson - Port Credit			1/3
32	Eastbound	Clarkson - Port Credit			3/5

The above Figures were derived from the basline data recorded in the morning of 1988 June 21 between 10:13 and 10:44. The start of the test zone was 600 wheel revolutions (1580 m) west of Port Credit Station.

6.4 VIBRATION ABSORBER

The vibration absorber was built according to UTDCI Drawing EX215-600-001. The design was based upon finite element analysis which helped to select a configuration which would exhibit a single vertical heave mode within the frequency range of interest (0 to 25 Hz). The test program for the absorber was carried out in two parts: a) a shop test to calibrate the device frequency versus applied mass and b) the actual road test of car 2030 with absorber installed.

6.4.1 Shop Test

MTO were asked to conduct the shop test on the absorber using the impact hammer method. The schematic Figure 13 shows the locations used in the test. Two locations (1 and 2) were used to record response level while locations 1 through 5 were used to excite the absorber.

Table 6.4-1 shows the measured mass of each of the weights used while Table 6.4-2 gives the measured frequencies for each test.

TABLE 6.4-1

Wt. No.	Mass - kg
*1	66.3
2	65.7
3	67.5
4	67.0
5	67.8
6	65.8
7	66.5
8	65.6
9	67.7
10	67.2
11	67.8

TABLE 6.4-2

Wts.	Response Location	Impact Location	Measured Absorber Frequency - Hz	Coherence	Phase Angle - Deg
1-5	1	1	12.75	1.0	-72.2
	1	2	12.75	1.0	-85.2*
	1	3	12.75	0.997	-84.3
	1	4	12.75	0.996	-91.3
	1	5	12.75	0.999	-79.1
	2	1	12.75	0.997	-70.0
	2	2	12.75	0.997	-83.0
	2	3	12.75	0.996	-78.6
	2	4	12.75	0.997	-83.2
	2	5	12.75	1.0	-75.5
1-7	1	1	11.5	0.997	-59.3
	1	2	11.5	1.0	-66.7
	1	3	11.5	1.0	-68.1
	1	4	11.5	0.997	-62.6
	1	5	11.5	0.997	-49.7
	2	1	11.5	0.995	-58.8
	2	2	11.5	0.997	-69.2
	2	3	11.5	0.997	-65.1
	2	4	11.5	0.999	-65.7
	2	5	11.5	—	—

TABLE 6.4-2 (Cont'd)

Wts.	Response Location	Impact Location	Measured Absorber Frequency - Hz	Coherence	Phase Angle - Deg
1-7	2	1	11.5	0.999	-62.1
	2	3	11.5	0.995	-60.4
	2	4	11.5	1.0	-69.1
	2	5	11.5	0.997	-47.8
	1	1	11.5	.996	-57.7
	1	3	11.5	.997	-65.1
	1	4	11.5	1.0	-64.4
	1	5	11.5	.997	-52.4
1-9	1	1	10.75	1.00	-70.3
	1	3	10.75	0.992	-72.0
	1	4	10.75	0.983	-79.4
	1	5	10.75	-	-
	2	1	10.875	Cable to hammer broken only got frequency response	
	2	3	10.875		
	2	4	10.876	0.996	-56.6
	2	5	10.875	0.999	-
1-11	2	1	10.5	0.997	-52
	2	3	10.5	0.997	-69.7
	2	4	10.75	0.990	-34.1
	2	5	10.75	0.999	-35.0
	1	1	10.625	0.995	-66.5
	1	3	11.625	0.996	-69.4
	1	4	11.75	0.998	-41.5
	1	5	11.75	0.993	-37.7

An example of a typical impact response at location 1 due to a hit at location 1 is given along with the transfer function identifying the resonant frequency in Figure 14.

This data was used to select the weight arrangements required to tune the absorber to the car frequencies.

6.4.2 Road Test

The absorber performance was assessed using the vertical acceleration channels to the centre of the upper deck (#3), lower deck (#1) and intermediate deck (#5). The raw data during a worst one minute passage through the Port Credit to Clarkson test zone was averaged and presented as a frequency spectrum of r.m.s. acceleration as in Figures 15 and 18. Next pairs of channels were compared in a transfer function plot including coherence and phase angle displays as in Figures 16 and 19. The ride data was further processed to give I.S.O. format for easier comparison of configurations as in Figures 17 and 20. Owing to the large amount of data collected and the limited time available, only the most relevant portions were reduced and these are presented here. Refer to MTO (J. Tessinga) for further information. The runs made during the road test are listed below:

Date	Test Description
88 June 15-20	Instrumentation Shakedown
88 June 21	Baseline Run - 1 Round Trip
88 June 21	Vibration Absorber (10 Wts) (Scarborough - Pickering)
88 June 21	Vibration Absorber (11 Wts) (Pickering - Oakville)
88 June 21	Vibration Absorber (8 Wts) (Oakville - Union) + stanchions Connected (Union - Scarborough)
88 June 21	Vibration Absorber (10 Wts) + Stanchions Connected (Guildwood - Pickering)
88 June 21	Vibration Absorber (11 Wts) + Stanchions Connected (Rouge Hill - Guildwood)
88 June 22	Vibration Absorber (11 Wts) + Stanchions Connected + Koni mod (Pickering - Oakville)

Generally, the lower deck is affected more than either the intermediate or upper decks by the absorber. The spectra in Figures 15 and 18 show that the absorber with eleven weights and tuned to 10.75 Hz reduces the lower deck response by a factor of 10 over a narrow (≈1 Hz) frequency band.

There is no damping designed into the absorber and thus its influence is restricted to this narrow band. There is a measurable reduction in the ride spectrum for the lower deck in Figure 17 and for the intermediate deck in Figure 18 and upper deck in both Figures.

Data from combined tests with stanchions and Koni mods was not reduced because insignificant improvement in ride was perceived although addition of stanchions did further quieten the upper deck relative to the other two decks. Should further analysis of this data be required, contact J. Tessinga at MTO.

6.5 SECONDARY DAMPER BUSH MOD

The secondary bush mod was installed on the Koni dampers of Car 2030 in the evening of 1988 June 21. The four bushings of each damper were chamfered to 5/16" as recommended for the optimum configuration (3-2) in Reference 4. Because both the stanchions and vibration absorber were already installed in the test car, this mod was tested in combination with the others first as shown in Table 6.5-1 below:

TABLE 6.5-1

Date	Activity	Direction	Location
1988 June 22	Recorded test of modified Koni bush + six stanchions in + vibration absorber with 11 weights	Eastbound	Union to Pickering
		Westbound	Pickering to Oakville
1988 June 22	Recorded test of modified Koni Bush + Six Stanchions IN	Eastbound	Oakville to Pickering
		Westbound	Pickering to Eglington
1988 June 22	Recorded test of modified Koni Bush only - 1 round trip	Westbound	Eglington to Oakville
		Eastbound	Oakville to Pickering
		Westbound	Pickering to Danforth

Two configurations were processed for ride quality. The first is the Koni bush mod alone which is shown in Figures 33 to 35 for channels 3 and 5 in terms of narrow band spectra, transfer function and ISO plot. The second, the Koni bush mod plus six stanchions connected is shown in Figures 36 to 38 for channels 1 and 3.

6.6 COMPARISON OF BILEVEL I, II AND III RIDE QUALITY

During this investigation, UTDCI had the opportunity with MTO and CCW help, to gather data not only on Bilevel I cars but also Bilevel II and III cars. Because the structural design had not significantly changed among the three marks, there was little change expected in the occurence of the dither problem.

The Bilevel I cars were measured in 1987 April in terms of ISO weighted acceleration in the initial ride tests documented in Volume I in Appendix B. When the question of selecting an appropriate test car for the Houdaille modified bushing test arose, a one day survey of seven Bilevel II Houdaille equipped cars was carried out on 1988 June 8. A short while later, acceptance tests were started by CCW on the first Bilevel III car at Willowbrook yard. Included in these were ride tests with and without a modified Houdaille damper bush installation. Table 6.6-1 below lists the measured ride on the upper deck of seven Bilevel II cars in terms of Leq in dB for 1 min. averaging time from a 2512 B&K Human Response vibration meter on 1988 July 8.

TABLE 6.6-1

Car #	1 Min. Leq dB	1 min. mean Acceleration r.m.s. g	ISO Ride RCB hrs.	Location
2101	133.0	0.0455	0.72	Mimico - Union
2102	130.5	0.0341	1.25	Scarborough - Danforth
2104	129.0	0.0281	1.78	Oakville - Clarkson
2105	131.0	0.0362	1.10	Port Credit - Clarkson
2107	131.5	0.0383	0.98	Hamilton - Burlington
2108	128.0	0.0256	2.10	Burlington - Oakville
2109	135.0	0.0573	0.43	Scarborough - Danforth

There was some dither present on all cars with car 2109 giving the strongest reading of all those tested.

In the case of Bi-level III cars, data taken on 1988 October 7 as part of CCW's acceptance test program is shown in Figures 39 to 41 inclusive. This data includes the effect of the Houdaille bush mod (4 serial bushes chamfered at 5/8") and indicates the presence of a dither vibration on the 12.5 Hz band (indicated by "x" on the plots) especially on the upper deck.

7.0 CORRELATION OF MODEL AND TEST RESULTS

The degree to which the test results and calculated trends agree are one measure of success in any program of this kind. In this section we will present a comparison between theory and test for each of the tested mods where the data permits.

7.1 STANCHION MOD

Recall that the stanchion mod was examined theoretically in the CCW finite element model (See Volume I, Section 5.3) only. Thus theoretical results are available for the relative effect of adding stanchions in terms of the two natural modes (12 Hz and 16 Hz). This data was used to derive a ratio of improvement at each frequency which was then able to be compared to the test results from (Reference 11, Volume I) and section 5.2 of this report. Table 7.1-1 shows the improvement.

TABLE 7.1-1

MODIFICATION		UPPER DECK ACC'N REDUCTION FACTORS			
		FEA [1]	CCW SHOP TEST	GO SHOP [3] TEST	GO ROAD [4] TEST
STANCHIONS	8 SQUARE	15 Hz 12 Hz	17 Hz	16 Hz	15 Hz 12 Hz
	6 STAGGERED	3.33:1 1.2:1	NA	NA	NA
	4 ZIG-ZAG	NA[2]	NA	4.8:1	3.3:1 1.2:1
	4 SQUARE	NA[2]	NA	4.0:1	1.27:1 1.5:1
		2.67:1 1.0:1	30:1	2.8:1	DELETED

NOTES:
1. Average of 4 nodes from centre on car centreline.
2. Not able to be modelled with ½ car model.
3. Average of 4 stations on car centreline at beam crossings either side of centre.
4. Based on 1 location at longitudinal centre of upper deck laterally offset to seat.

The above table shows that the FEA model predicts more improvement for the 15, 16 Hz upper deck mode than for the 12 Hz body bending mode as seen in the test data. Even though direct comparison between all tests and configurations is not possible, the trends agree with respect to increasing the number of stanchions (an improvement of from 1.3:1 to 3.3:1 in the 15 Hz mode response with 4 and 8 stanchions).

The predicted frequencies of the two modes also agree closely with both the Go shop test and road test.

7.2 VIBRATION ABSORBER MOD

The vibration absorber design and test activities were supported theoretically by both the MSC/PAL2 F.E.A. program and the vertical dynamics program Bilevel 5. During design, the natural modes of the absorber were predicted for two mass settings and one of these is compared with the MTO impact hammer verification test results. Figure 42 shows the test versus calculated agreement for the first vertical beam mode (the most important one from the design point of view). The predicted frequencies of this mode are higher than test values by about 1 Hz. This is likely because the actual wall thickness and/or beam dimensions are smaller than assumed in the model. However, the agreement is good and shows the correct trend with increasing mass.

When the absorber was tested on Car 2030, the ride was measured in terms of 1/3 octave band vertical r.m.s. g level and compared with the predicted values from model Bilevel 5. In the road test, the absorber had essentially low to zero damping and thus should be compared with the 2% critically damped case in Figure 43. This plot indicates a close agreement of the trend with absorber mass and shows that the predicted absorber vibration reduction is optimistic (38% predicted and 35% measured).

7.3 SECONDARY BRUSH MOD

The ride test data from the Koni secondary bush mod was analysed at two conditions: with and without the modification, and, then compared with earlier predicted performance. Because measurements of bushing stiffness were not made, the predicted values of bush stiffness from a linear PAL2 simulation were used. Figure 44 shows

the predicted ride on the upper and lower decks with bush stiffness. These were arrived at by matching the initial ride tests on Car 2030 (1987 April) with the model (Bilevel 6). The measured upper deck ride for the unmodified bush (link stiffness = 1.6 MN/m) is lower than the predicted value because the general level of ride improved between the 1987 April and 1988 June tests (See Figure 11). The second measured point with the modified bush installed (link stiffness = 0.54 MN/m) shows that some improvement has occurred but not enough to meet expectations. Further softening is necessary, and, in retrospect the 5/16" chamfer should have been made 5/8". The preload should also be reduced whilst a scheme is incorporated to prevent the washers either side of the bushings from slapping the link and making noise. Further tests would reveal the practicality of this and determine if rubber fatigue is critical.

7.4 IMPROVED DEFINITION OF PROBLEM TEST

The refined model Bilevel 8, described in Vol. I, Section 4.5 was matched in part to the test data from the improved definition test. Unfortunately, this tuning activity resulted in overly stiff chevrons (37 x) and overly soft sidebearers (5.8 x). These values were required to achieve simulation of both the 17 Hz Frame bounce and 12 Hz bolster bounce modes. Since the 17 Hz mode did not affect the area above the frame, it was decided to eliminate the 17 Hz mode from the simulation. This gave values for chevron and sidebearer stiffness which were closer to the original Dofasco values. The 17 Hz frame mode was not well defined by this test and is likely to have been a result of track input at this frequency and some form of structural bending of the frame. Another possible explanation is that the chevrons at one end of the truck were older and hence stiffer, than those at the opposite end causing a 17 Hz bounce at the stiff end only. Further test work could reveal the true behaviour of this mode.

8.0 CONCLUSIONS AND RECOMMENDATIONS

8.1 PHASE I TESTS

A two phase program to define the behaviour by test of the Bilevel I and II cars has been carried out. Phase I consisted of an:

a) initial test to survey 18 vehicles for vertical ride quality.

b) A test to evaluate four stanchion arrangements.

c) decision not to test the upper deck floor panel modification and primary truck dampers.

Data from all tests was recorded on mag tape and is accessible for further analysis through MTO, Transportation and Technology Branch, Downsview, Ontario in care of J. Tiessinga.

8.2 PHASE II TESTS

Phase II consisted of three tests including:

a) Improved definition of problem test which focussed upon the influence of the truck rigid body behaviour as it influences dither vibration.

b) Evaluation of a vibration absorber on the lower deck to reduce the first body bending mode response.

c) Evaluation of a modified Koni bushing arrangement (similar to one tested earlier by CCW in 1980) to reduce transmissibility of the secondary suspension in the area of 12 Hz.

Agreement between predicted and test results was mixed. For the stanchions, the reduction of the upper deck mode at 16 Hz was well supported by the finite element model results which showed the correct trend with increasing numbers of stanchions and which showed little improvement of the 12 Hz body bending mode from stanchions.

The vibration absorber design frequency variation with applied mass was in good agreement, within 1 Hz, of the predicted values from the PAL2 program. The discrepancy is mainly due to differences between

actual versus assumed beam cross-sectional shape. The ride measured with the absorber agreed quite well with the predicted levels, giving a measured 35% reduction in dither vibration compared to 38% predicted from the model.

The secondary bush mod results were correlated on the basis of a single test using measured ride results with estimated bushing stiffness. The agreement with the simulation results was not close because the general level of ride had improved since the model results were generated. The fact that the modified bushing stiffness was above the optimum by fifty times could explain why the improvement obtained was small.

A refined model describing the bi-level car was obtained from the test data. However, owing to anomalies in the required suspension values, the best match to the test data was not used and in its place, one with more physically realizable values of sidebearer and chevron stiffnesses was substituted (model Bilevel8). Confirmation of the above approach could only be made by testing individual truck rubber components.

8.3 RECOMMENDATIONS

From the testing carried out in phases I and II, it is recommended that:

a) The secondary damper bush mod be investigated further as a relatively inexpensive fix to the dither problem.

b) The refined model, Bilevel 8, be confirmed by validating sidebearer and chevron suspension values and defining the dynamic structural deflection of frame and bolster.

c) The stanchion mod be considered in conjunction with another modification which addresses the first body bending mode.

REFERENCES

1. J. Tiessinga, "Preliminary Report on the Test of Bilevel II Coach #2132 at Willowbrook", MTO Report BL III - 740, 1986 January 26.

2. UTDC, "Proposal for a GO Transit Bi-Level III Ride Quality Improvement Program", Proposal submitted 1987 April.

3. J.A.Johnston, Letter Proposal to Dr. T. AppaRao of MTO regarding tasks 3a,b,c and d, submitted 1988 April 8.

4. H. Pool, "Bi-level Damper Bushing Stiffness", CCRI Test Report #178, 1980.

5. D.C. Gilmore, "Bi-level III Subjective Ride Test Plan to Evaluate Stanchion Option, TP 230 600 002 1988 February 10.

6. D.C. Gilmore, "A New Solution Option for Bi-Level Dither", memo to J.A. Johnston, UTDCI/2290/87, 1987 July 14.

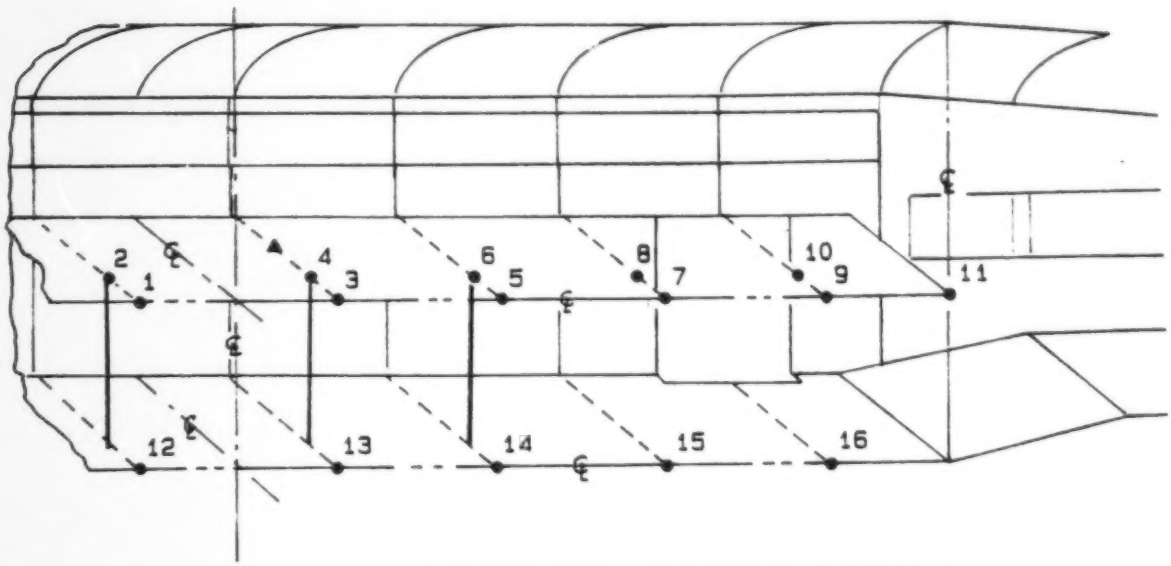


Figure. 1 Stanchion Mod Test Hammer Test Impact/
Response Locations

Bi-Level Ride Test

Name: _____ Date: _____

Affiliation: _____

Previous Involvement in Subjective Ride Tests: _____

Instructions: When asked to rate ride, place an "X" along scale opposite appropriate Test # and Zone #. The position of "X" should be where you think the ride of the current car is relative to the reference car.

Test #	Run #	ZONE	DIRN	Rating of 2030 vs 2001									Comments
				+4 Better	+ 3	+2	+1	0	-1	-2	-3	-4 Worse	
1.													
	1	1	WB										
	2	2	WB										
	3	2	EB										
	4	1	EB										
	5	3	EB										
	6	3	WB										
2.													
	1	1	WB										
	2	2	WB										
	3	2	EB										
	4	1	EB										
	5	3	EB										
	6	3	WB										

Figure. 2 Sample Passenger Response Form

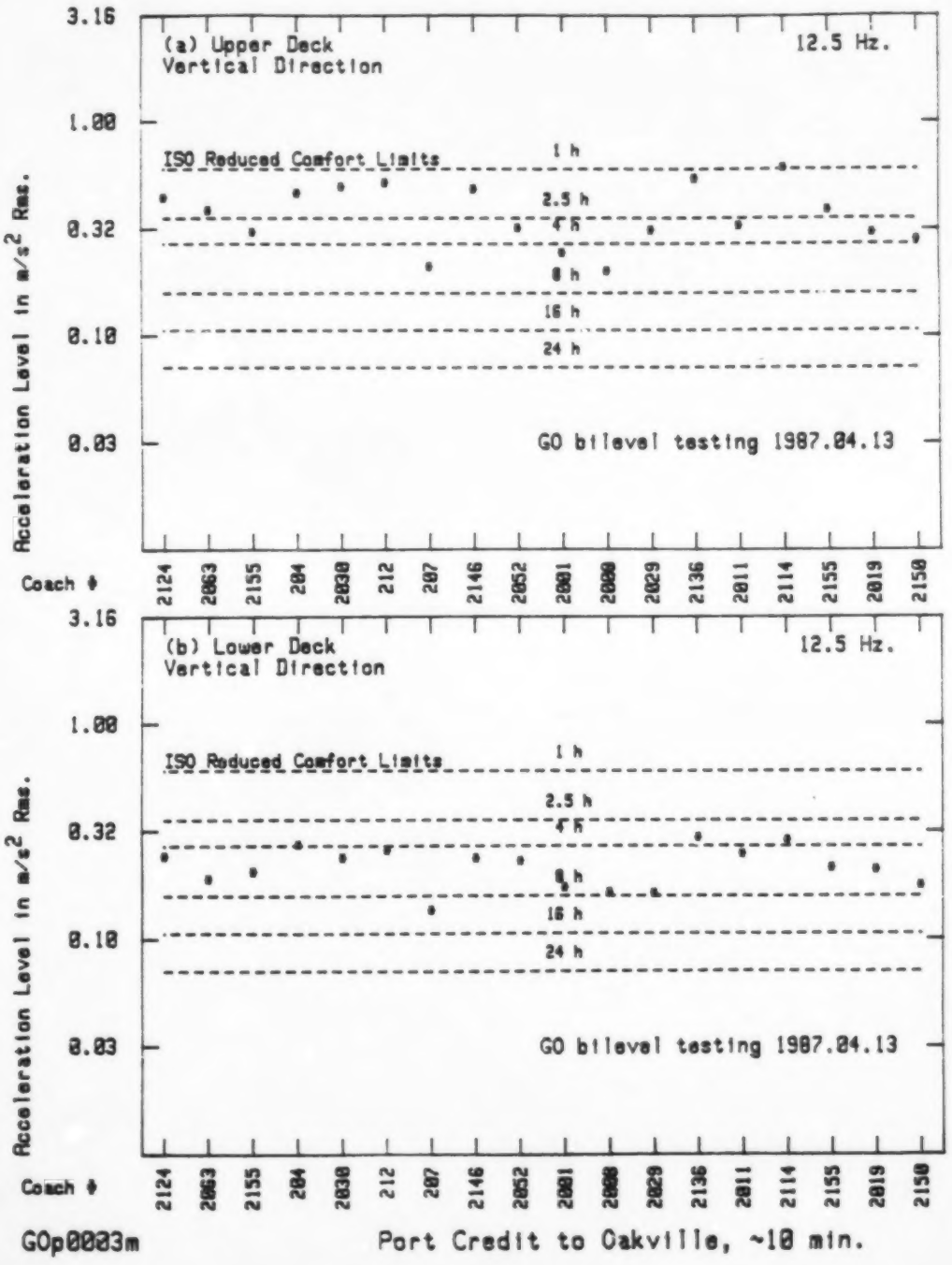


Figure. 3 Initial Ride Survey

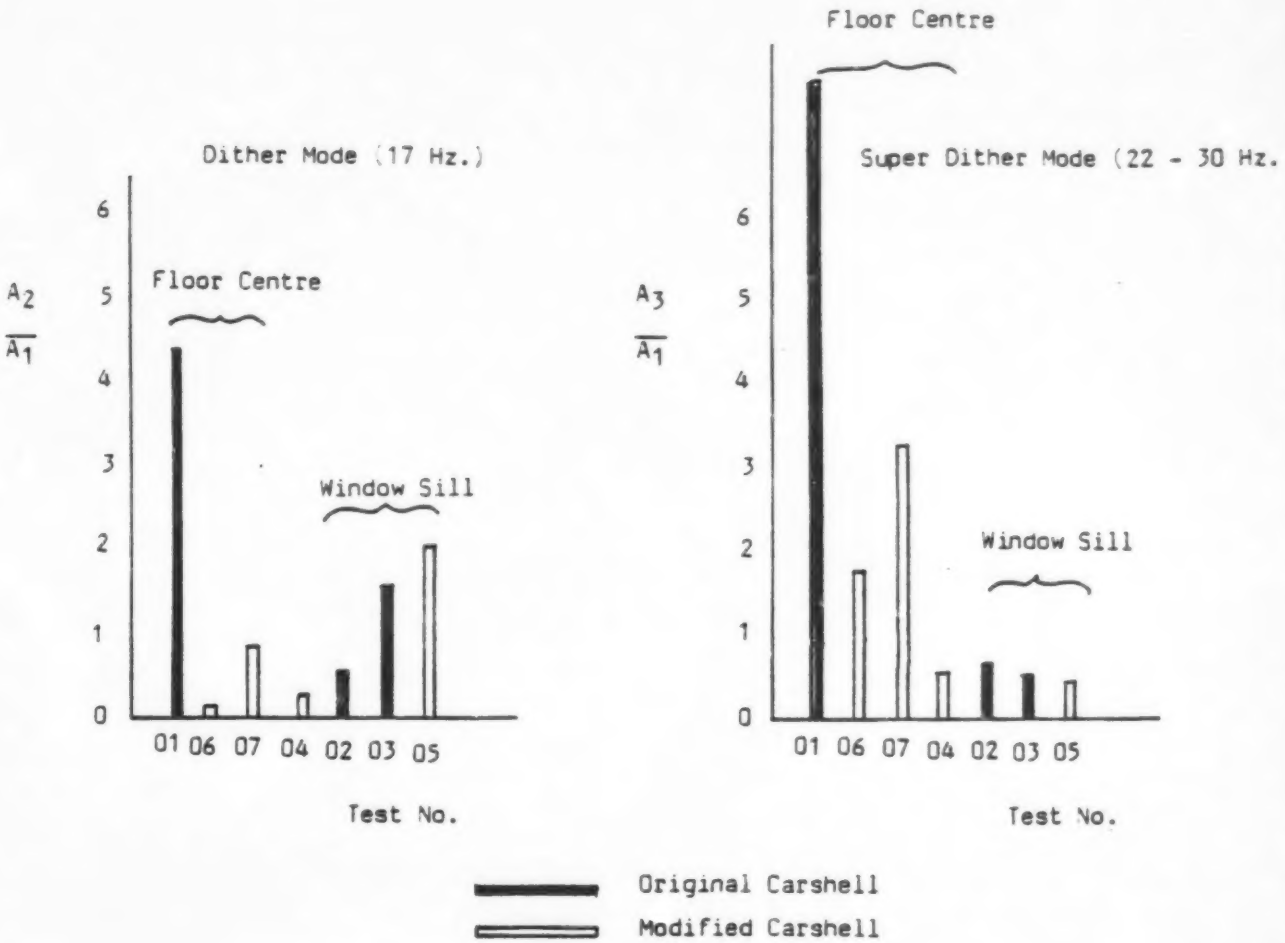


Figure. 4 Stanchion Mod Test: CCW Tire Drop Test on Bare Carshell

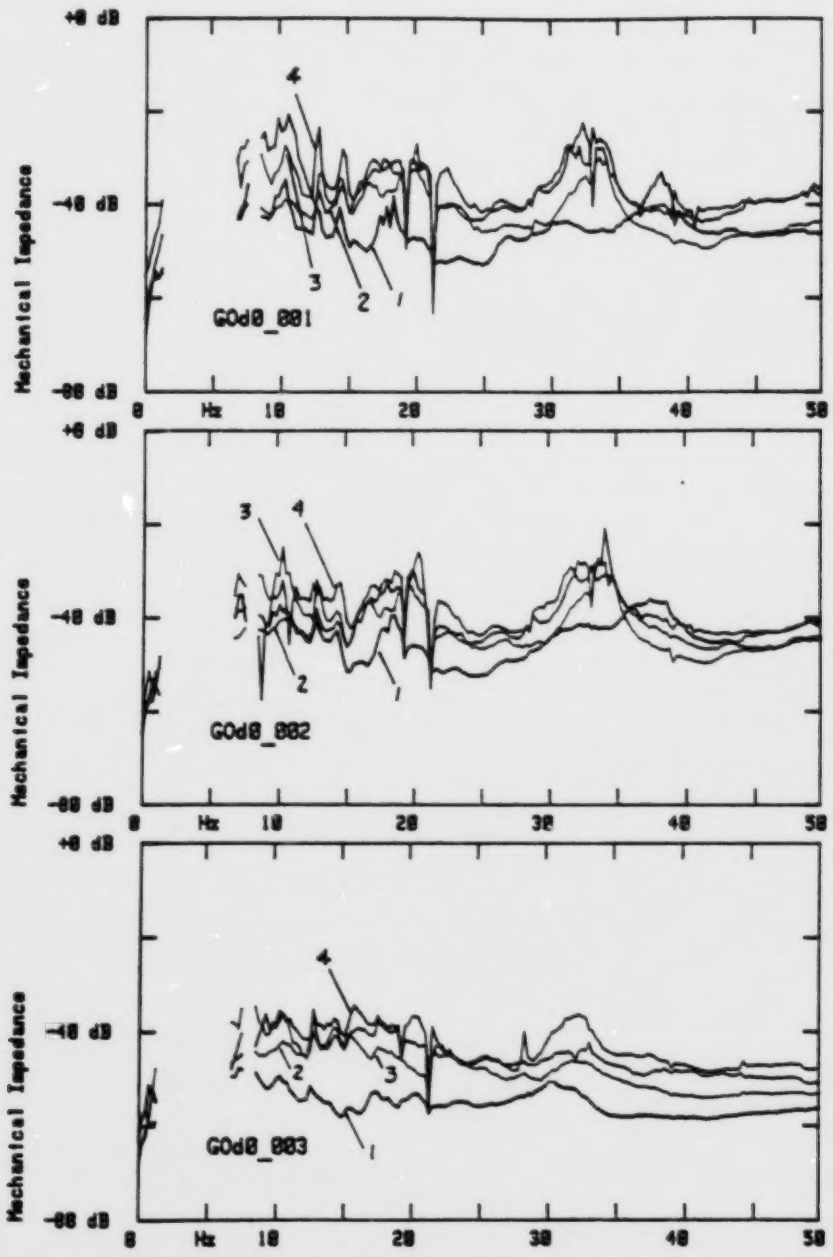


Figure 5 / Comparison of Coach Vertical Mechanical Impedances
1987.12.02, Coach 2030

TTh_01b, G0f003

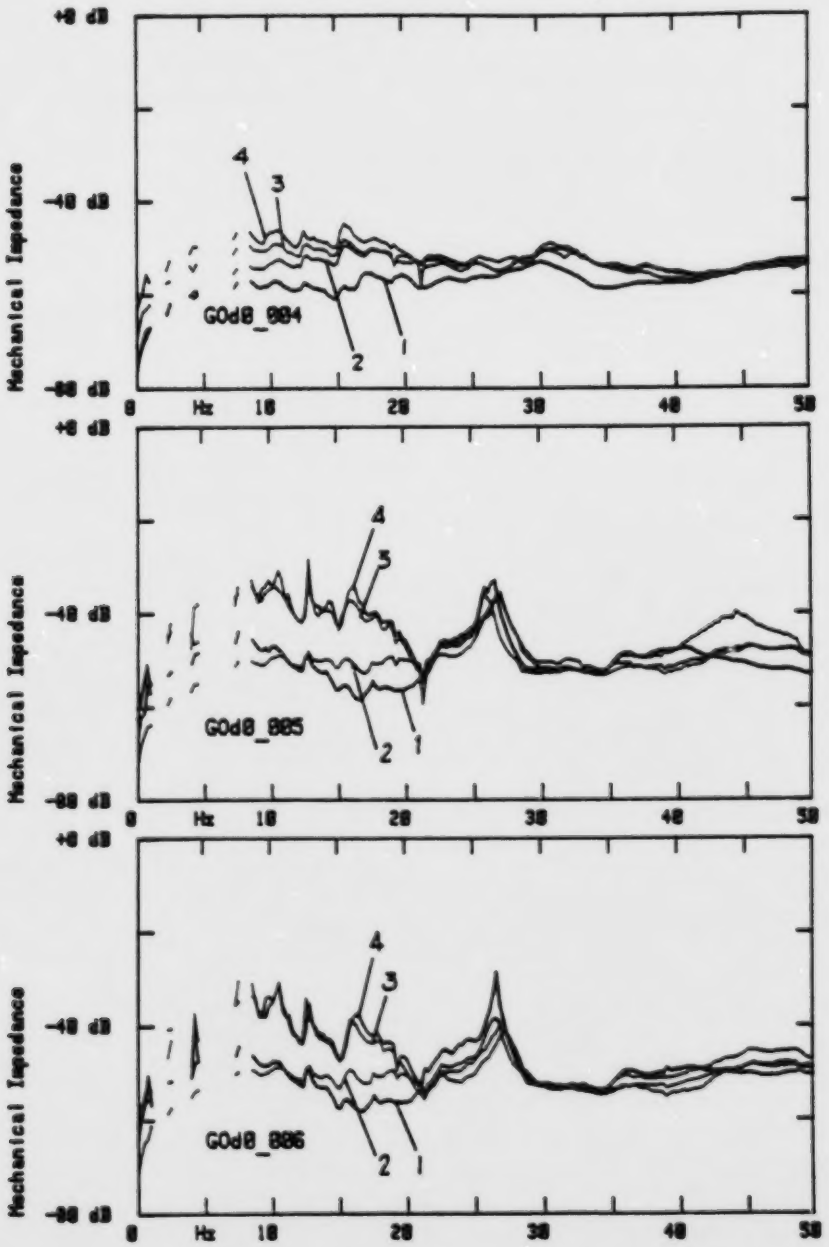


Figure 6 / Comparison of Coach Vertical Mechanical Impedances
1987.12.02, Coach 2030

TTh_01b, G0f003

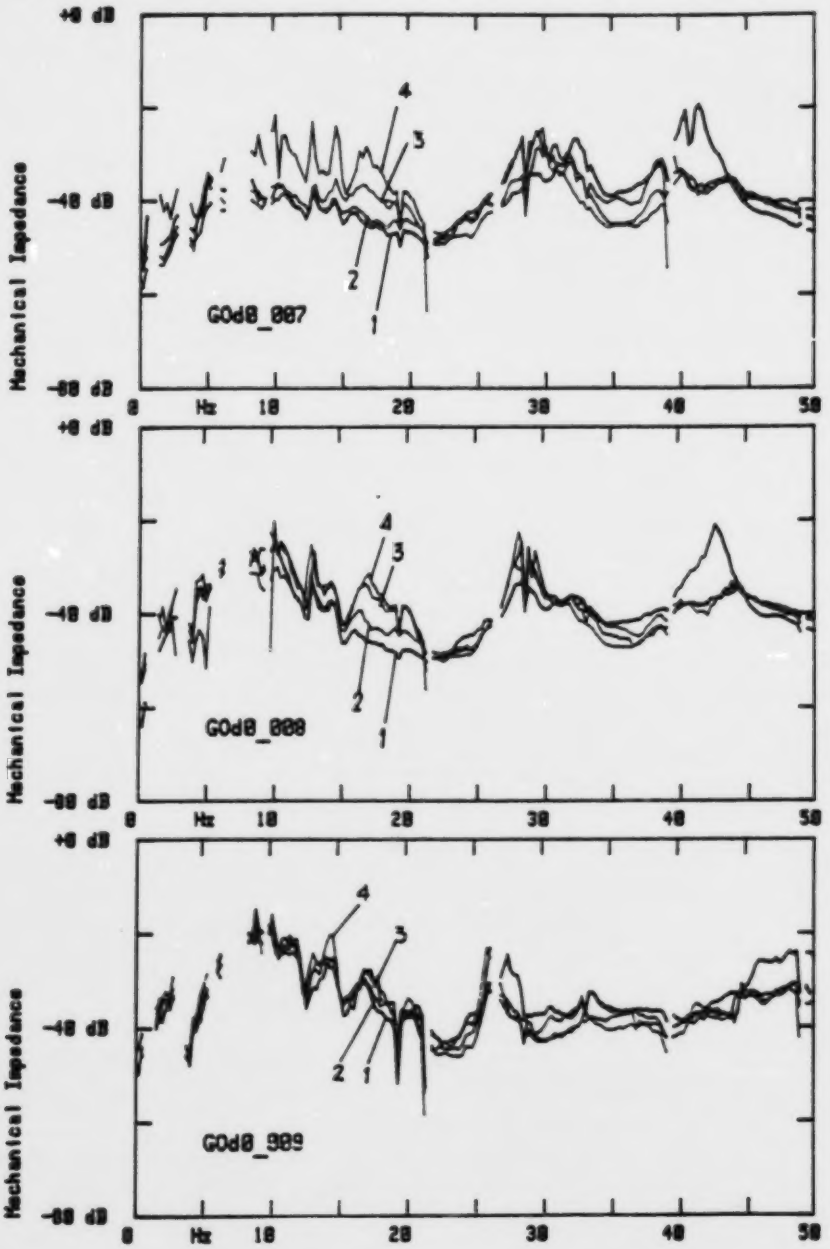


Figure 7 / Comparison of Coach Vertical Mechanical Impedances
1987.12.02, Coach 2030

TTh_01b, G0f003

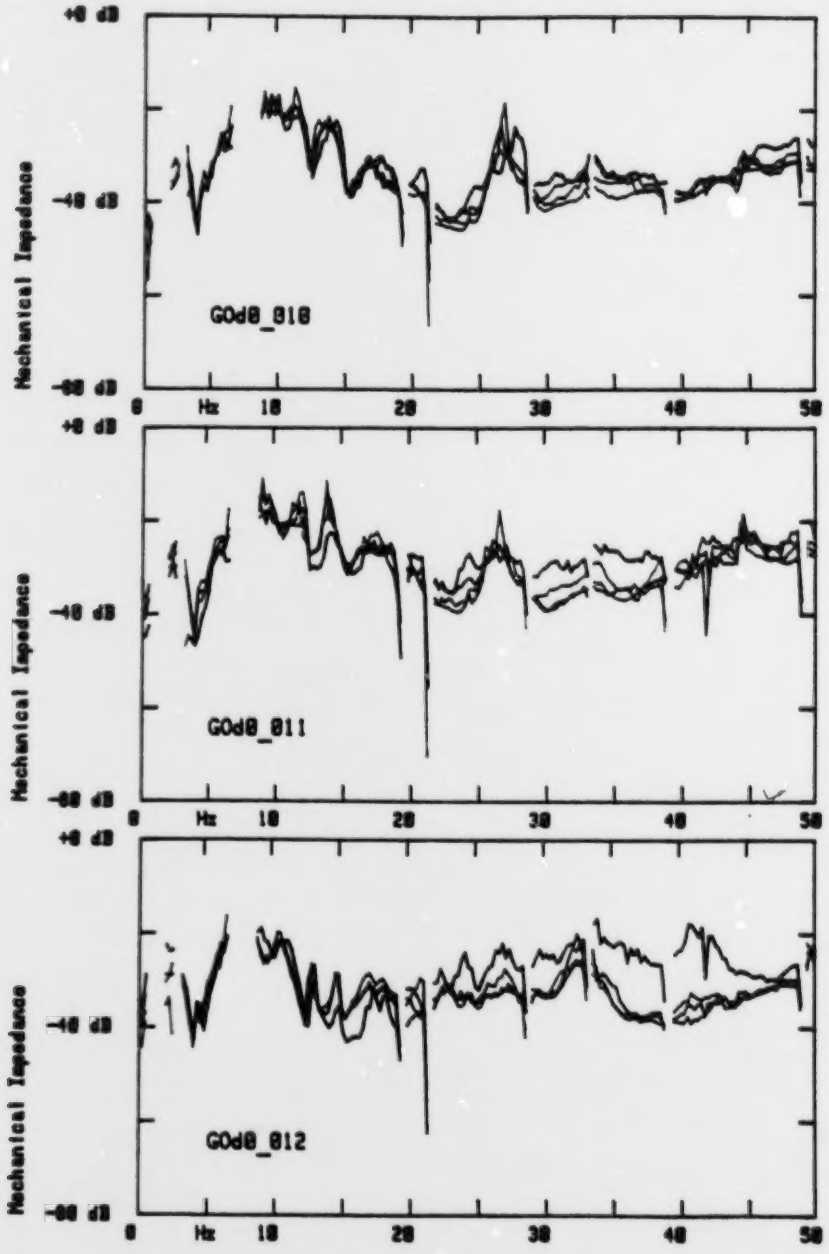


Figure 8 Comparison of Coach Vertical Mechanical Impedances
1987.12.02, Coach 2038

TTh_01b, G0f003

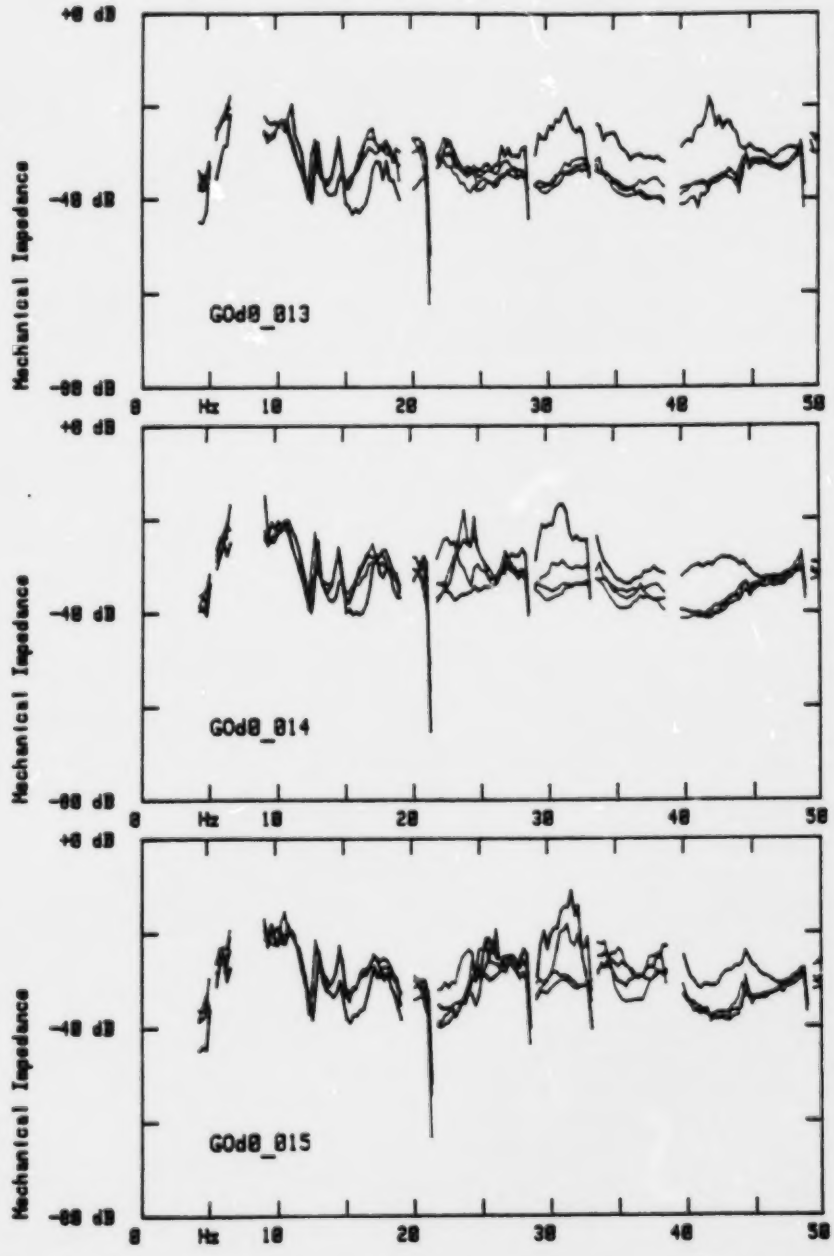


Figure 9 / Comparison of Coach Vertical Mechanical Impedances
1987.12.02, Coach 2038

TTh_01b, GOf003

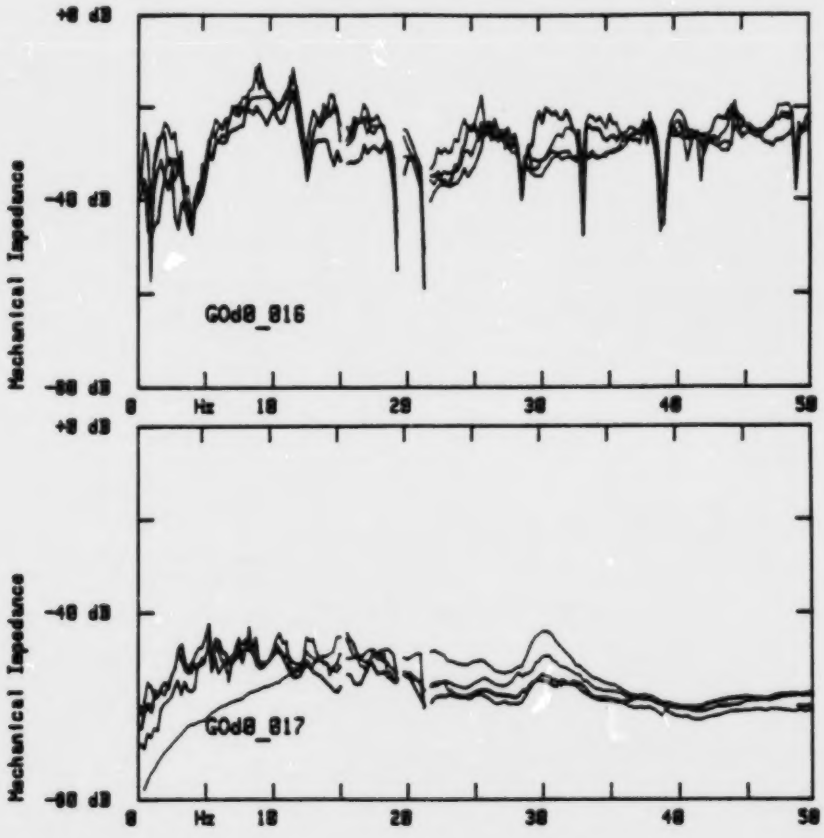


Figure10 / Comparison of Coach Vertical Mechanical Impedances
1987.12.02, Coach 2030
TTh_01b, GOf003

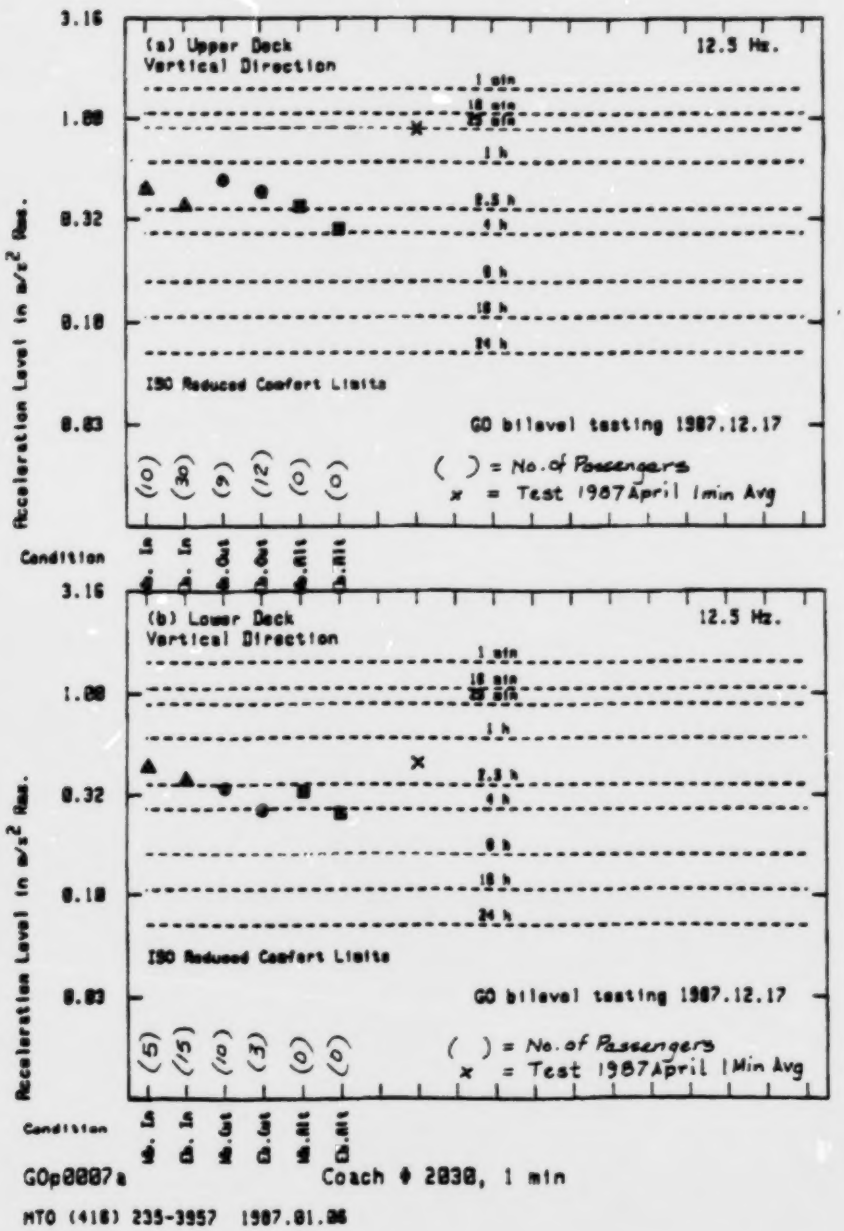
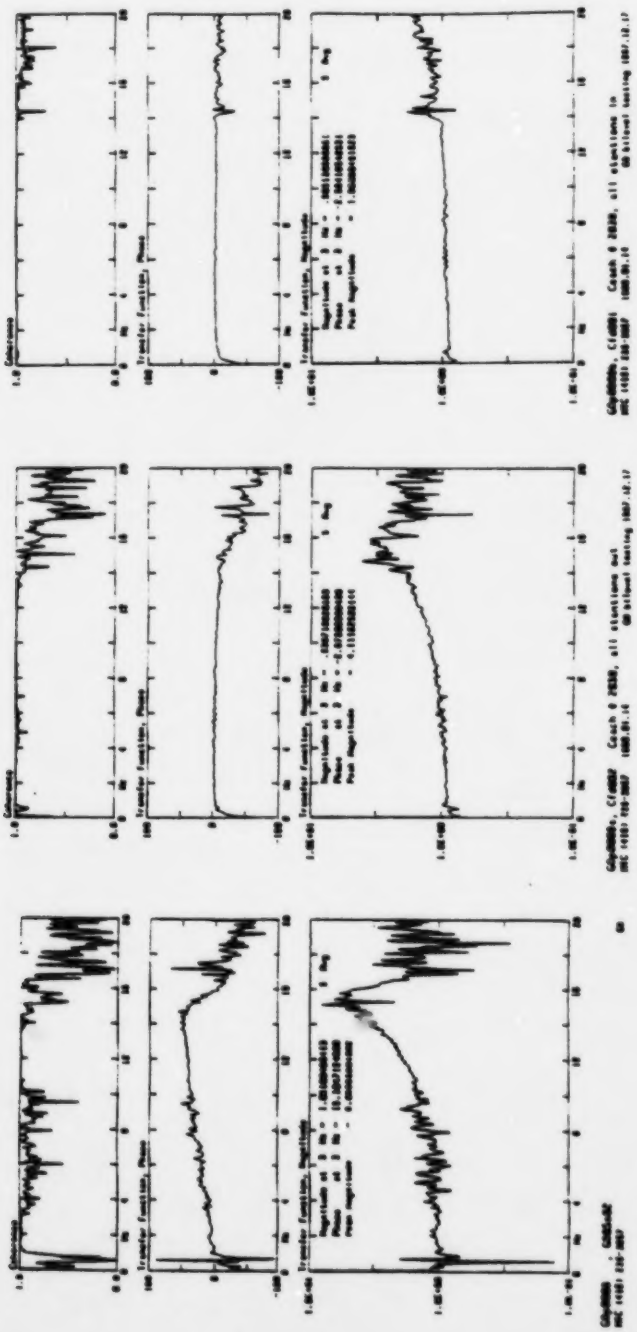


Figure. 11 Stanchion Test - Summary of Ride Results



12a 12b 12c

Figure. 12 Stanchion Test: Measured Transfer Functions

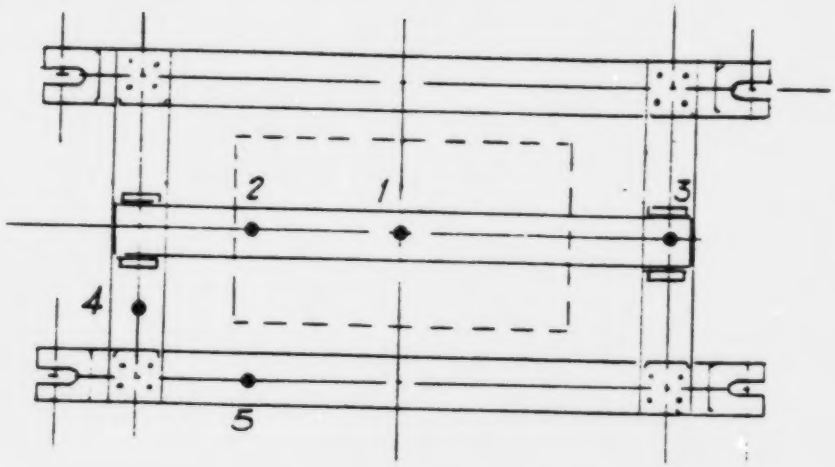


Figure. 13 Vibration Absorber: Hammer Test Locations

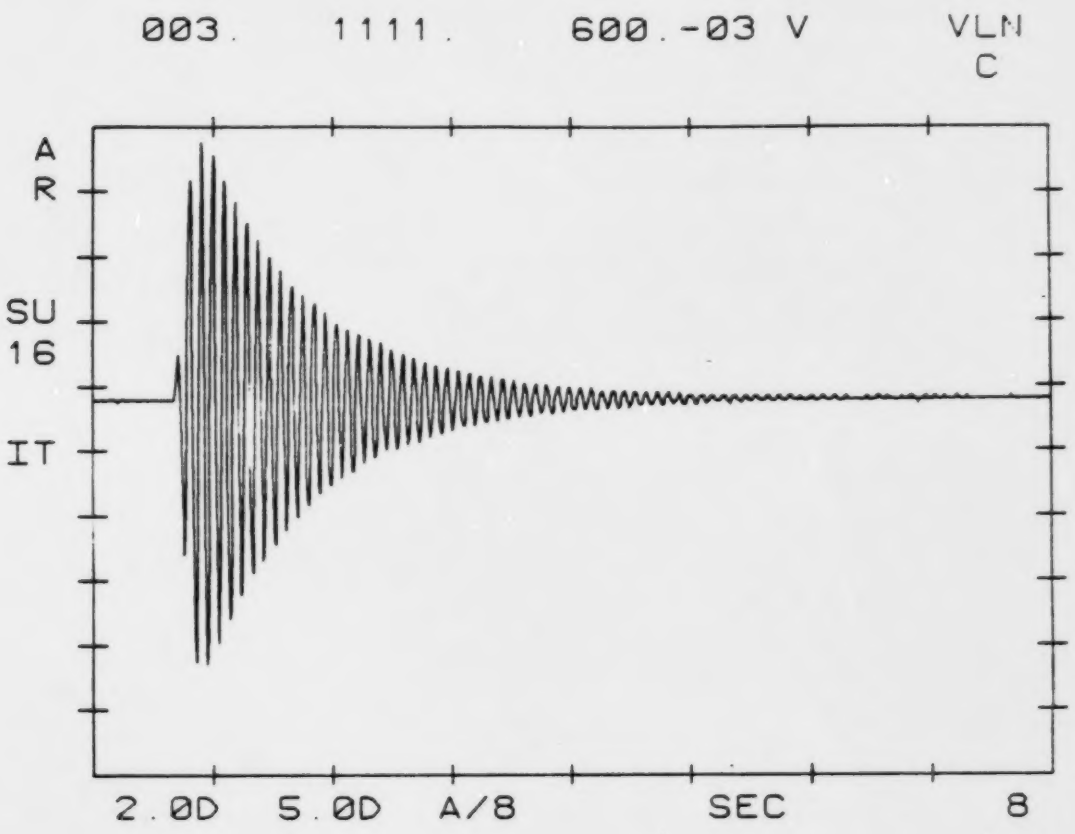


Figure. 14a Vibration Absorber: Decay Plot of Response at 1 to Hammer Impact at Location 1

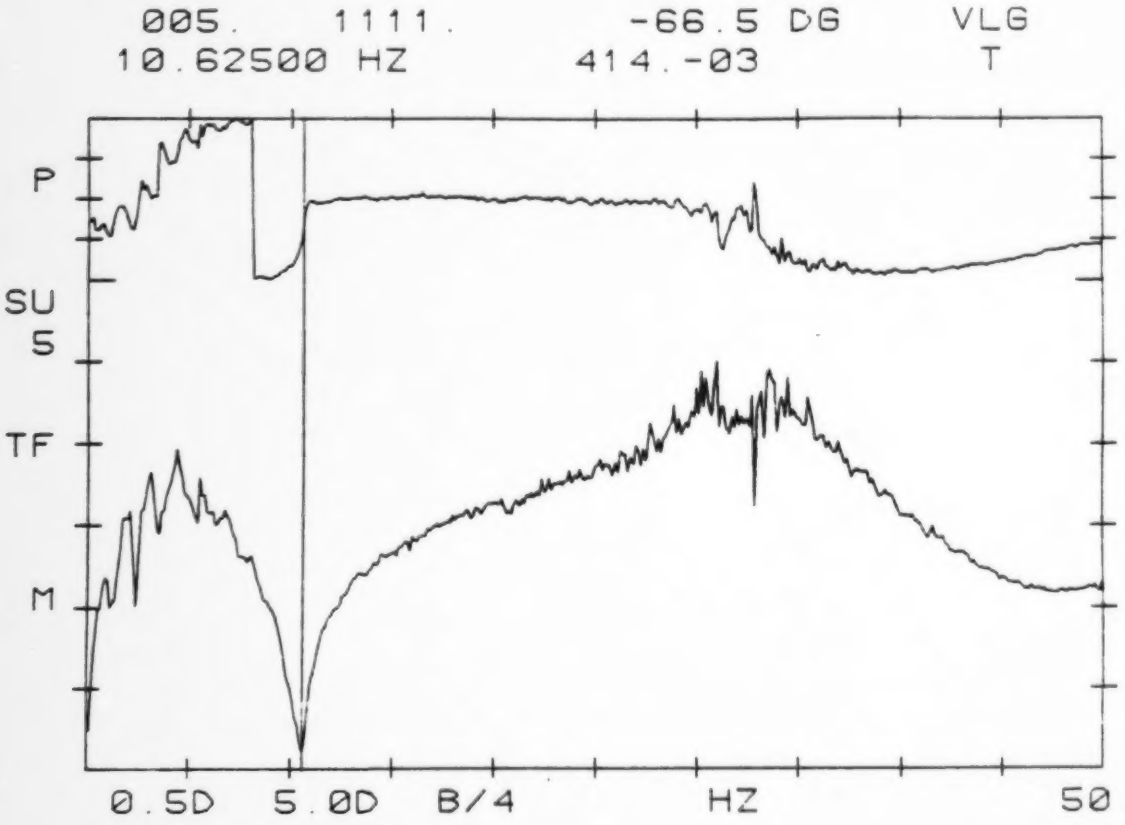
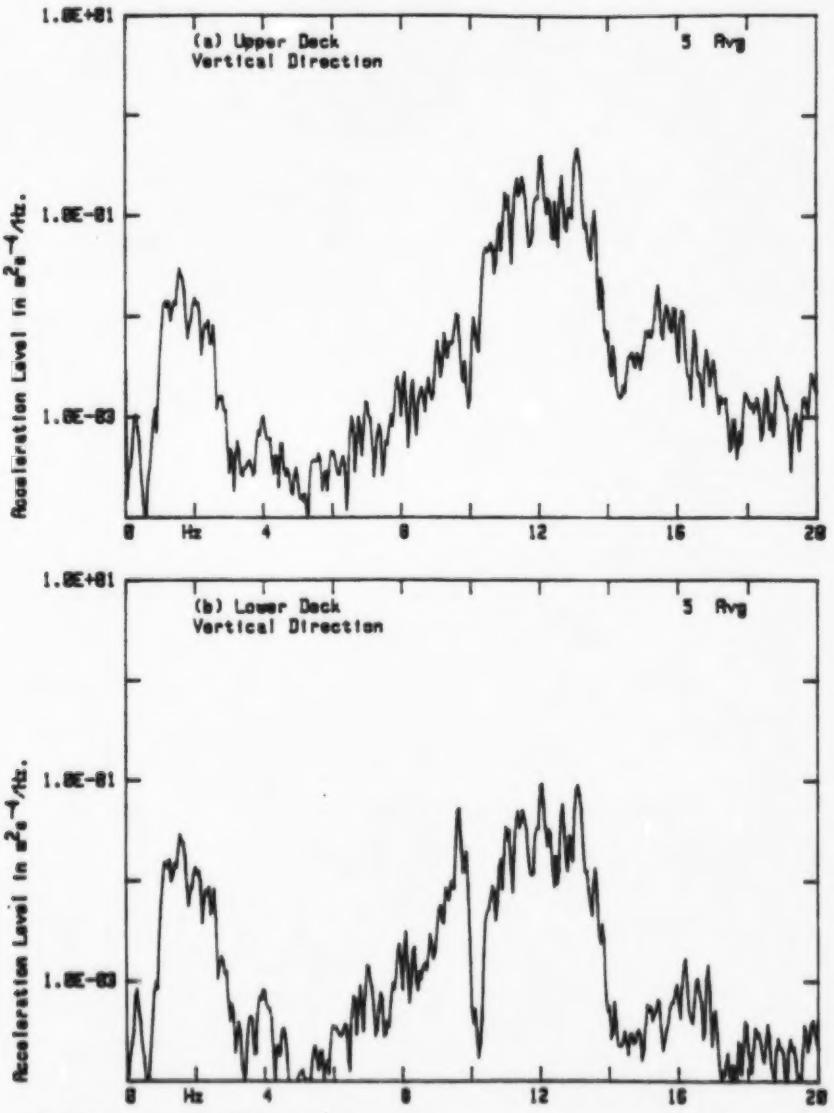


Figure. 14b Vibration Absorber: Impedance
Measured at Location 1



GOp0001d, GOk_010304
GO Transit testing 1988.05.21
Channel A is Channel 01, Channel B is channel 03, Port Credit to Clarkson
13:15:14 to 13:16:14, Eleven weights on the vibration absorber
MTC (416) 235-3957 1988.09.22

Figure. 15

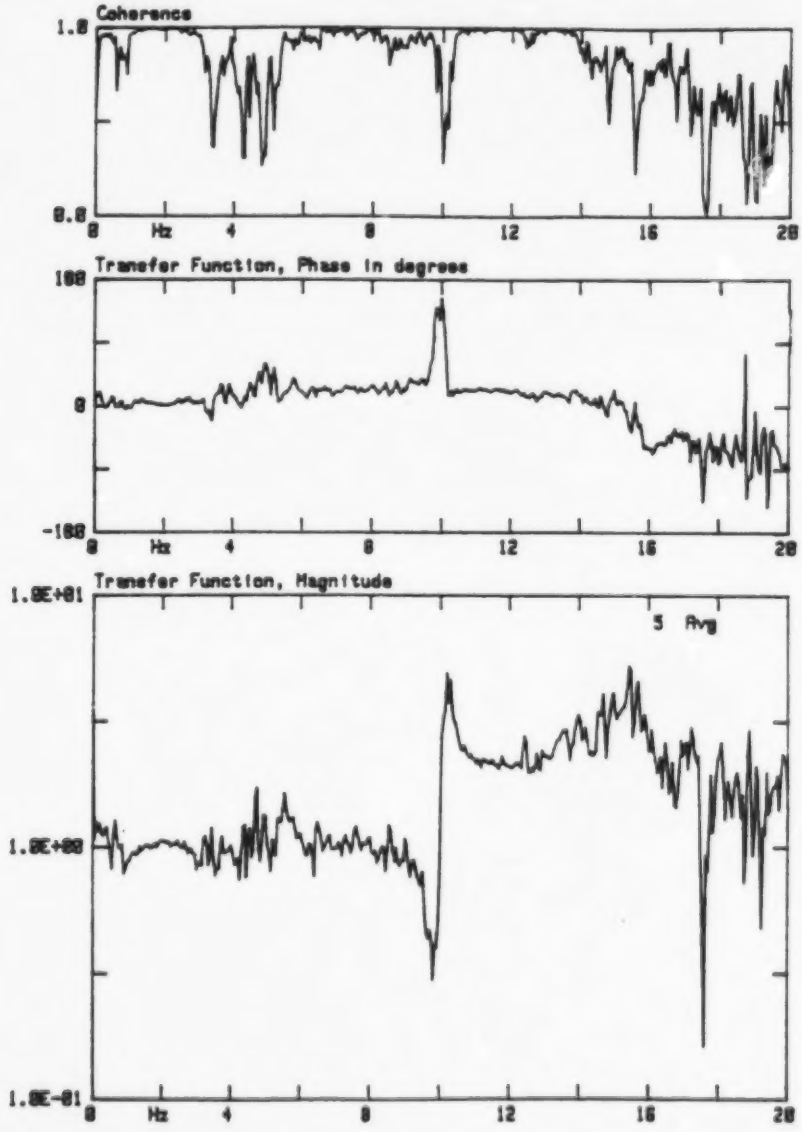
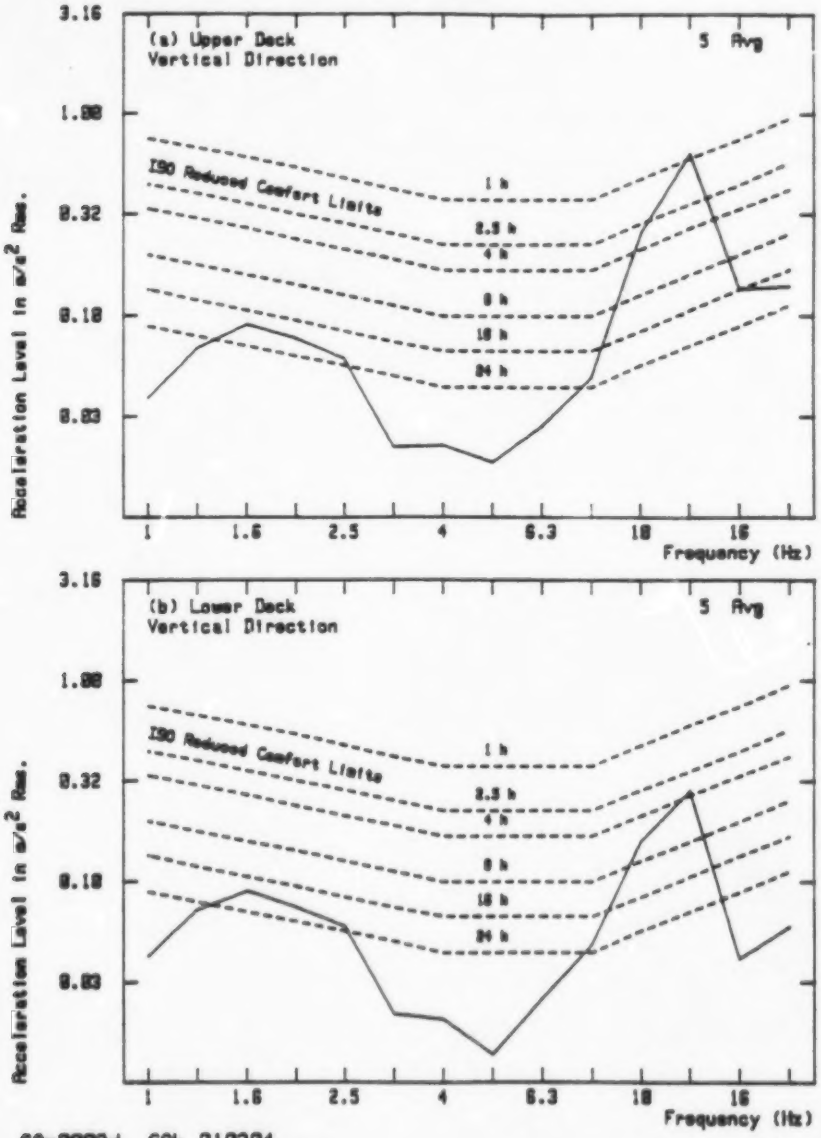
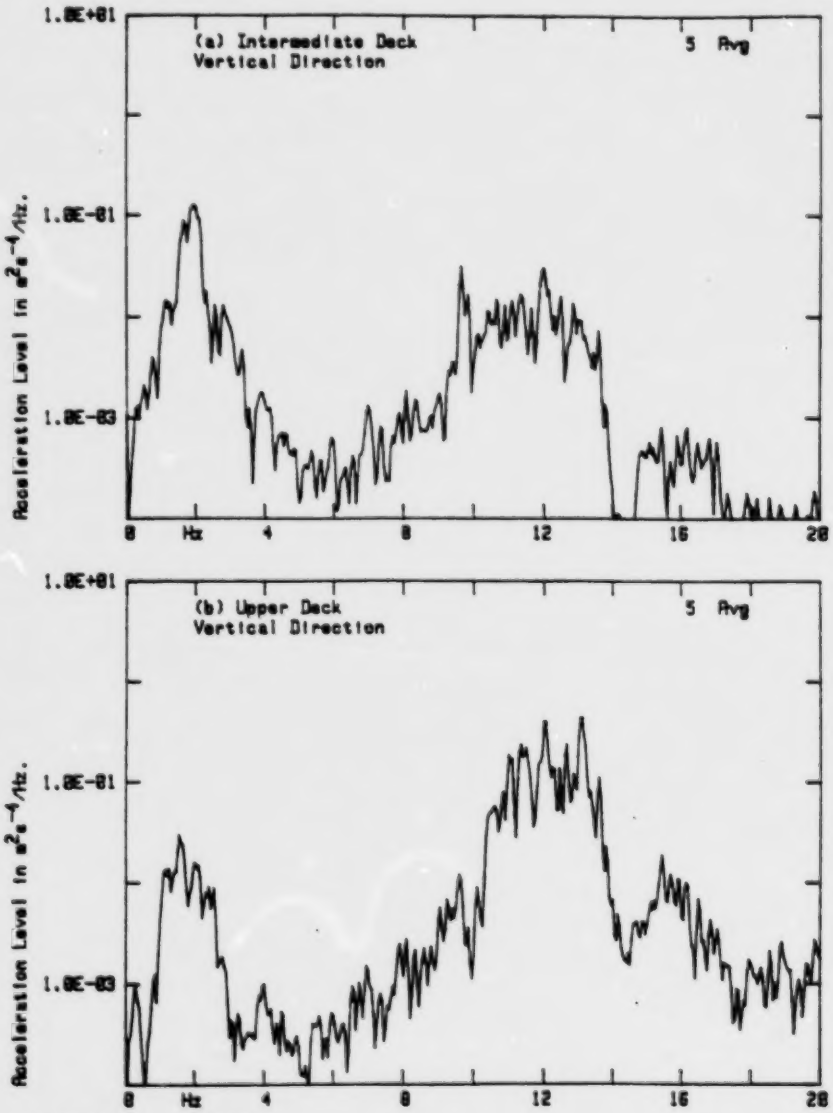

Coherence
1.0
0.0
0
Hz
4
8
12
16
20
Transfer Function, Phase in degrees
180
0
-180
Transfer Function, Magnitude
1.0E+01
5 Avg
1.0E+00
1.0E-01
GOp0008d , GOk_010304, Transfer Function between Upper & Lower Deck
GO Transit testing 1988.05.21
Channel A is Channel 01, Channel B is channel 03, Port Credit to Clarkson
13:15:14 to 13:16:14, Eleven weights on the vibration absorber
MTC (416) 235-3957 1988.05.22

Figure. 1-



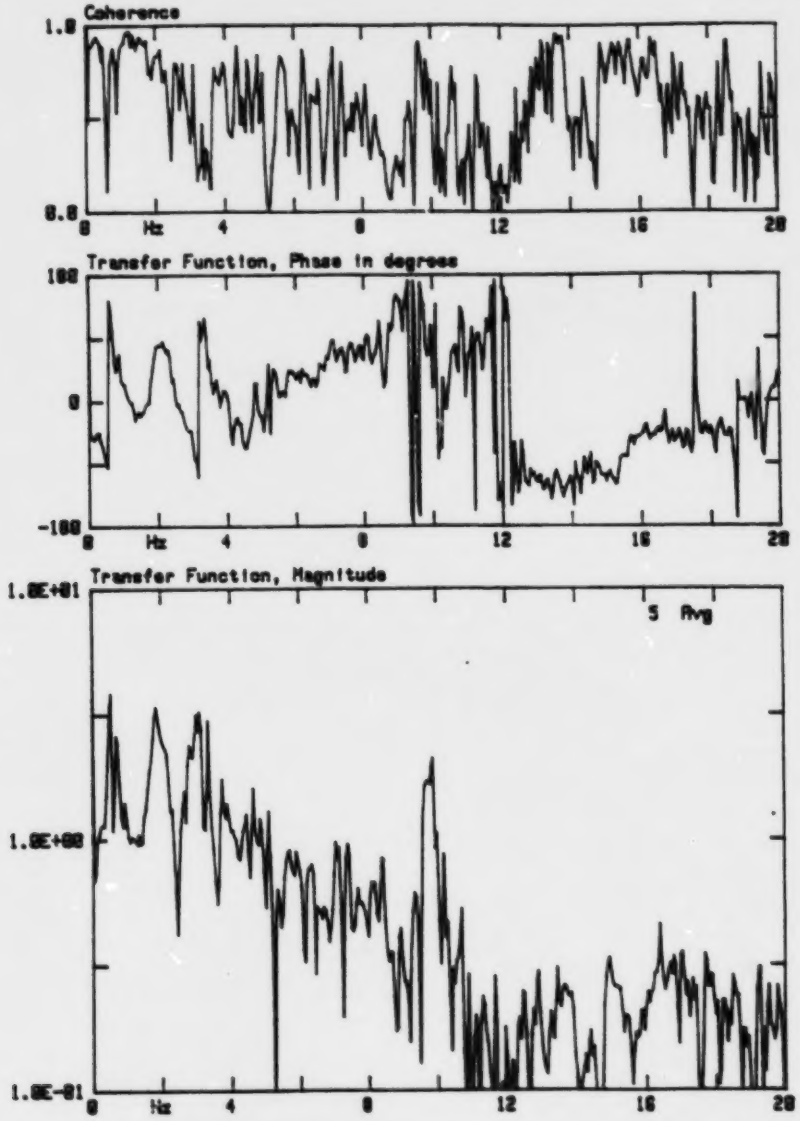
GOp0002d, GOk_010304
GO Transit testing 1988.06.21
Channel A is Channel 01, Channel B is channel 03, Port Credit to Clarkson
13:15:14 to 13:16:14, Eleven weights on the vibration absorber
MTC (416) 235-3957 1988.09.22

Figure. 17



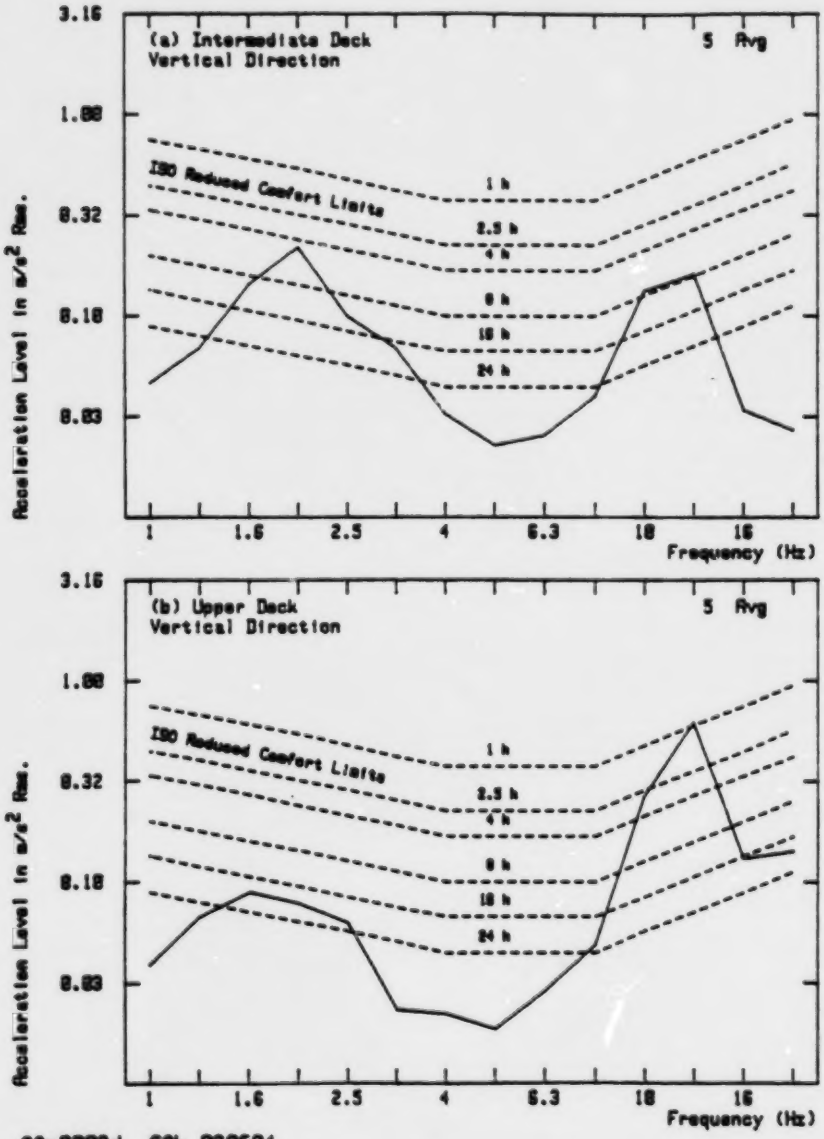
GOp0001d, GOk_030504
GO Transit testing
Channel A is Channel 03, Channel B is channel 05, Port Credit to Clarkson
13:15:14 to 13:16:14 , June 21, Eleven weights on the vibration absorber
MTC (416) 235-3957 1988.09.22

Figure. 18



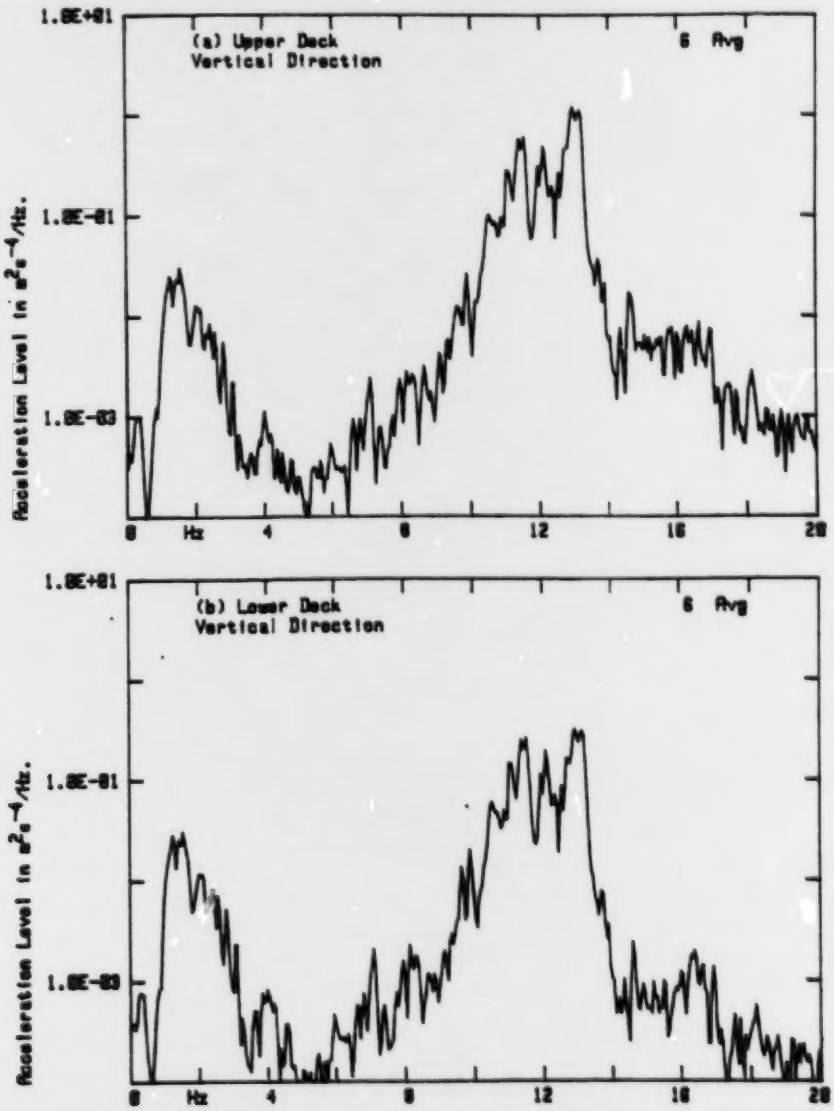
GOp0008d , GOk_030504psfer Function between Intermediate & Upper Deck
GO Transit testing
Channel A is Channel 03, Channel B is channel 05, Port Credit to Clarkson
13:15:14 to 13:16:14 , June 21, Eleven weights on the vibration absorber
MTC (416) 235-3957 1988.09.22

Figure. 19



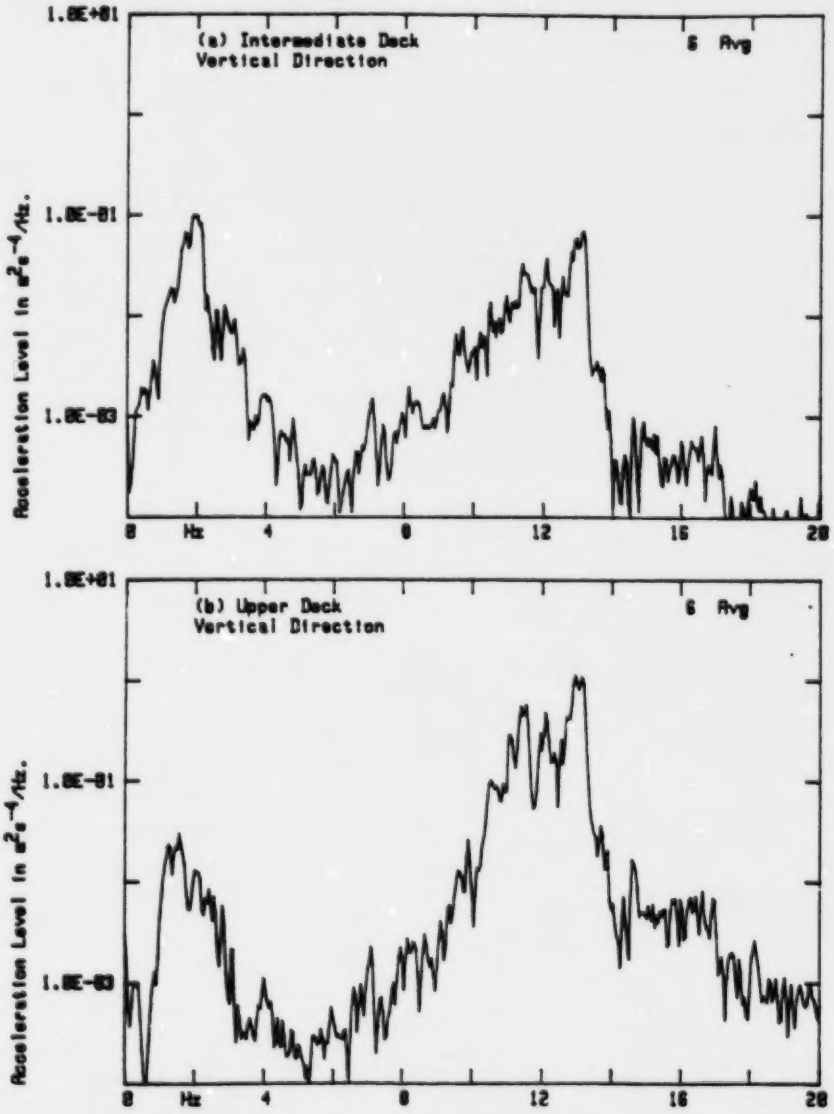
GOp0002d, GOk_030504
GO Transit testing
Channel A is Channel 03, Channel B is channel 05, Port Credit to Clarkson
13:15:14 to 13:16:14 , June 21, Eleven weights on the vibration absorber
MTC (416) 235-3957 1990.09.22

Figure. 20



GOp0001d, GOk_010301
GO Transit Testing 1900.00.21
Channel A is Channel 01, Channel B is Channel 03, Port Credit to Clarkson
10:13:43 to 10:14:45, Baseline Data.
MTC (416) 235-3957 1900.00.22

Figure. 21



GOp0001d, GOk_230501
GO Transit Testing 1988.06.16
Channel A is Channel 03, Vert. Acc. UD, Channel B is Channel 05, Vert. Acc. ID,
Starting time 203.18.13.45 to 14.45 about 600 revolutions out Port Credit
MTC (416) 235-3957 1988.09.22

Figure. 22

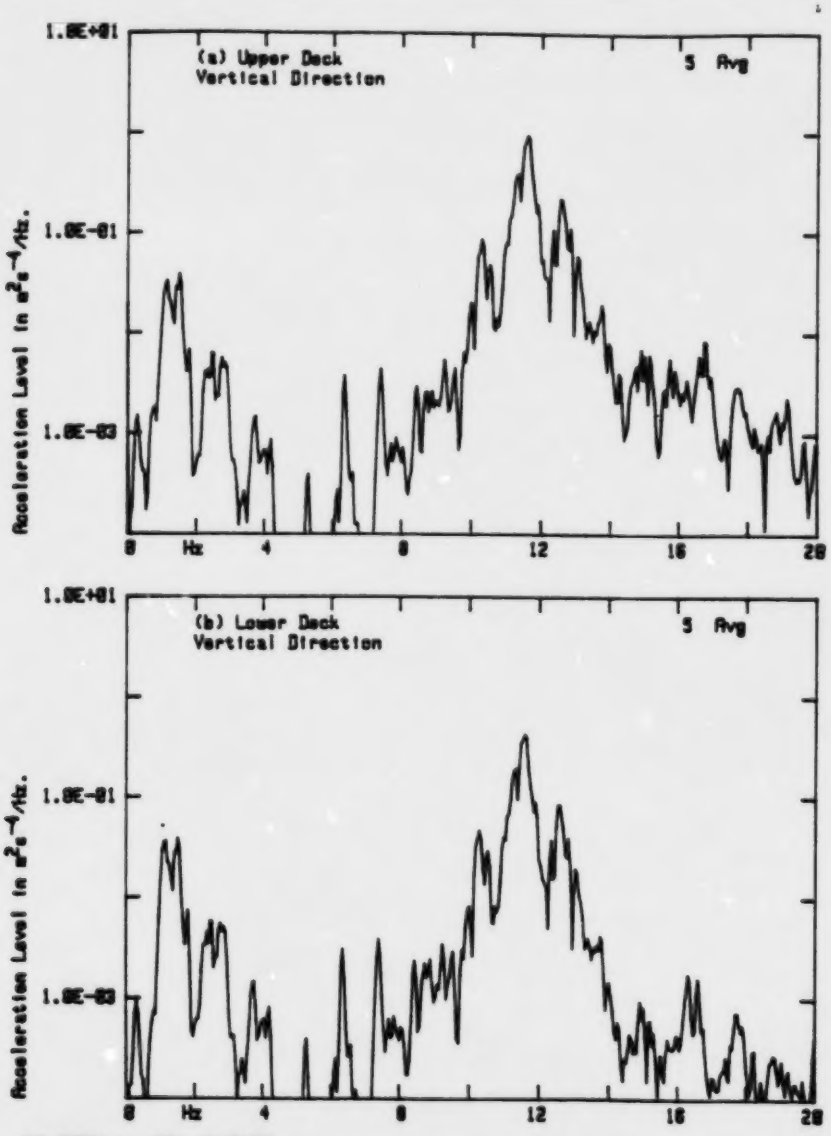

1.0E+01
(a) Upper Deck
Vertical Direction
5 Avg
1.0E-01
1.0E-03
Acceleration Level in m²s⁻⁴/Hz.
0 Hz 4 8 12 16 20
1.0E+01
(b) Lower Deck
Vertical Direction
5 Avg
1.0E-01
1.0E-03
Acceleration Level in m²s⁻⁴/Hz.
0 Hz 4 8 12 16 20
GOp880ld, GOk_818382
GO Transit testing 1988.06.21
Channel A is channel 01, Channel B is channel 03, Clarkson to Port Credit,
18:40:40 to 18:41:40, Baseline Data
MTC (416) 235-3957 1988.08.22

Figure. 23

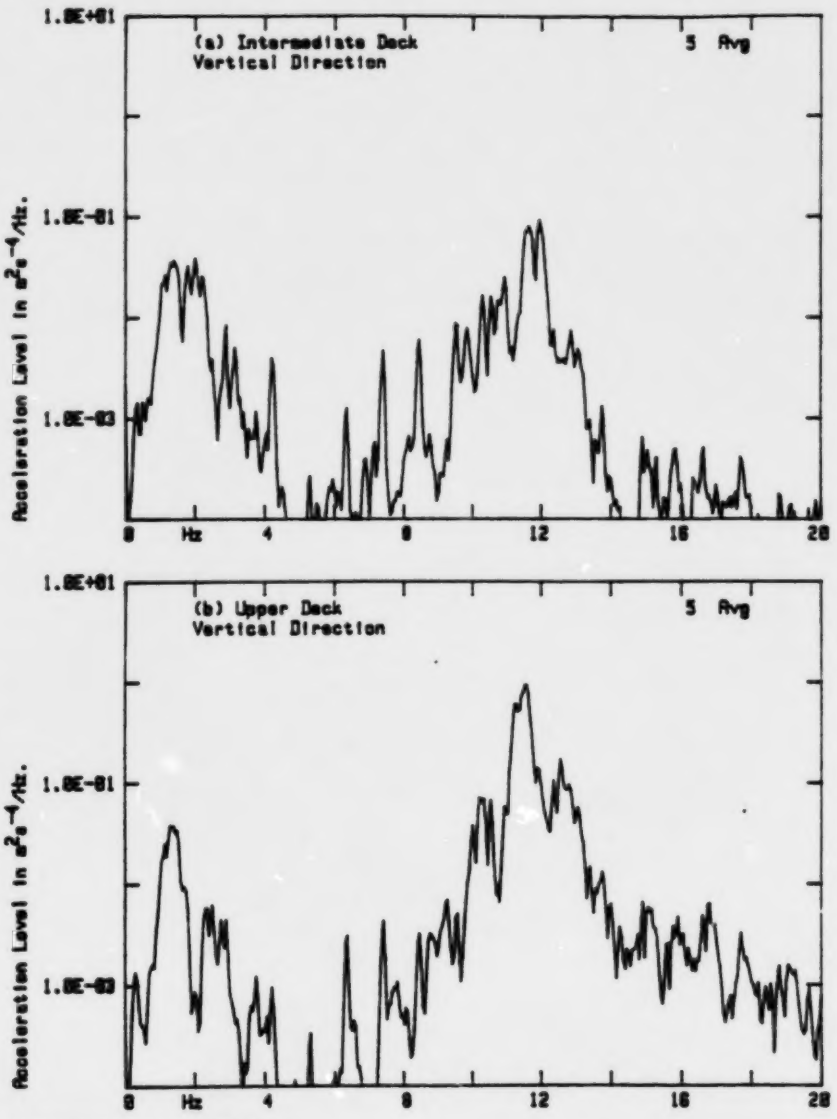
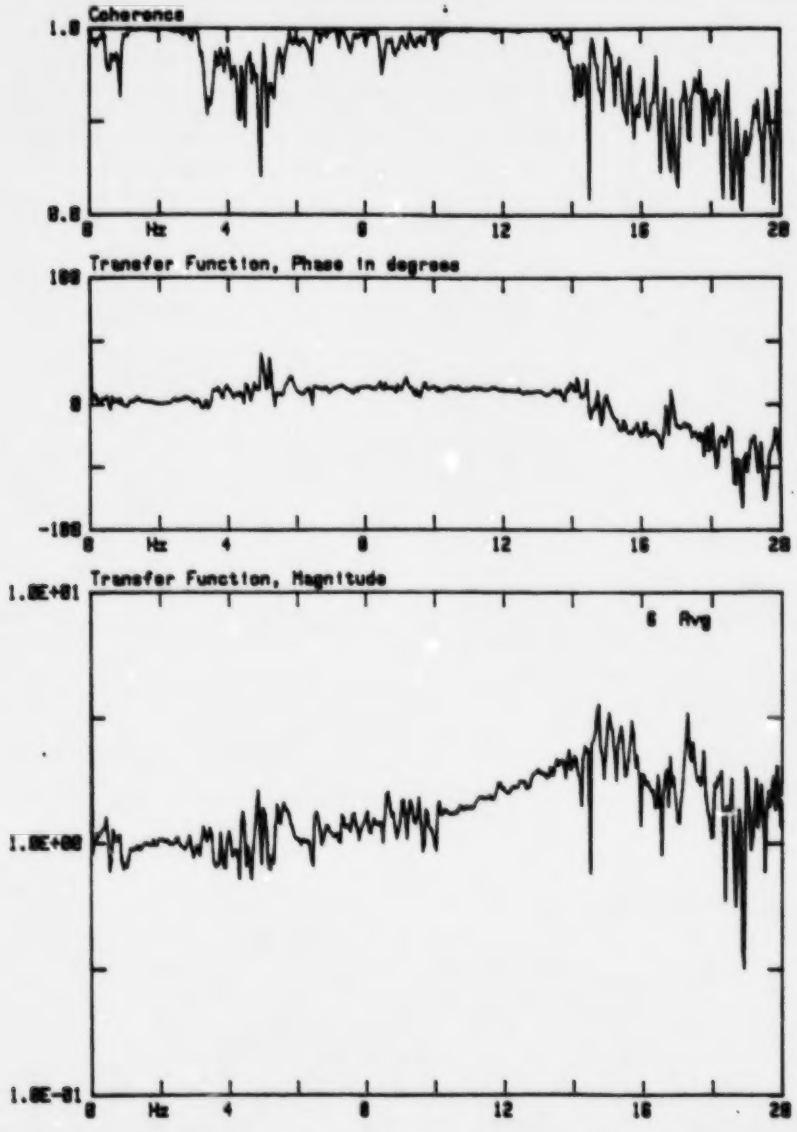
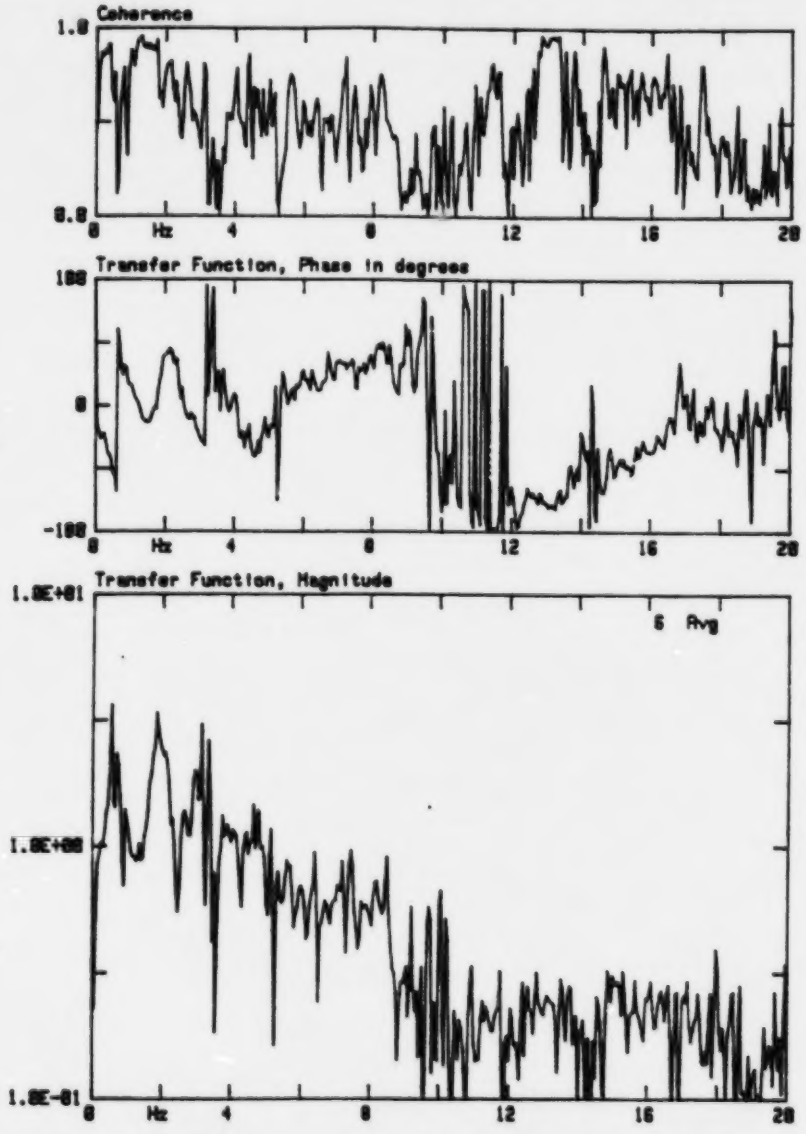

(a) Intermediate Deck
Vertical Direction
5 Avg
1.0E+01
1.0E-01
1.0E-03
Acceleration Level in g²e⁻⁴/Hz.
0 Hz 4 8 12 16 20
(b) Upper Deck
Vertical Direction
5 Avg
1.0E+01
1.0E-01
1.0E-03
Acceleration Level in g²e⁻⁴/Hz.
0 Hz 4 8 12 16 20
GOp0001d, GOk_030502
GO Transit testing 1988.06.21
Channel A is channel 3, Channel B is channel 5, Clarkson to Port Credit,
18:40:40 to 18:41:40
MTC (416) 235-3957 1988.09.22

Figure. 24



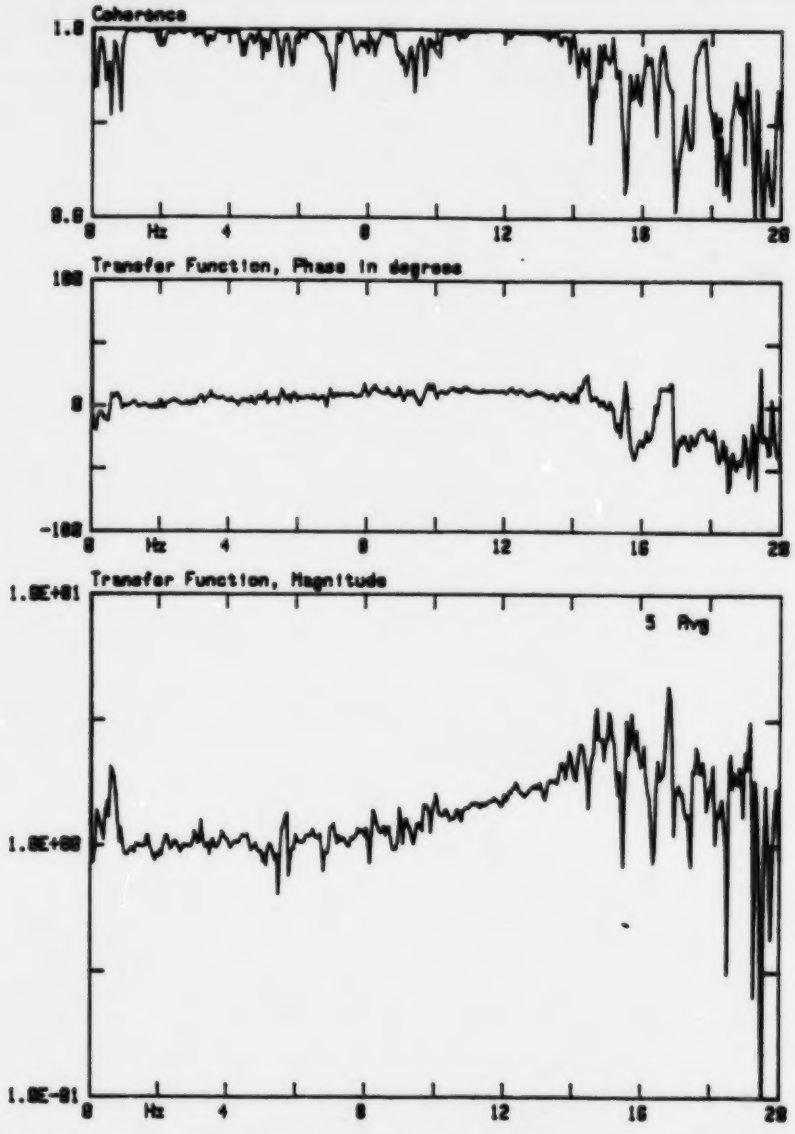
GOp8008d , GOk_818381, Transfer Function between Upper & Lower Deck
GO Transit Testing 1988.08.21
Channel A is Channel 01, Channel B is Channel 03, Port Credit to Clarkson
18:13:45 to 18:14:45, Baseline Data.
MTC (416) 235-3957 1988.09.22

Figure. 25



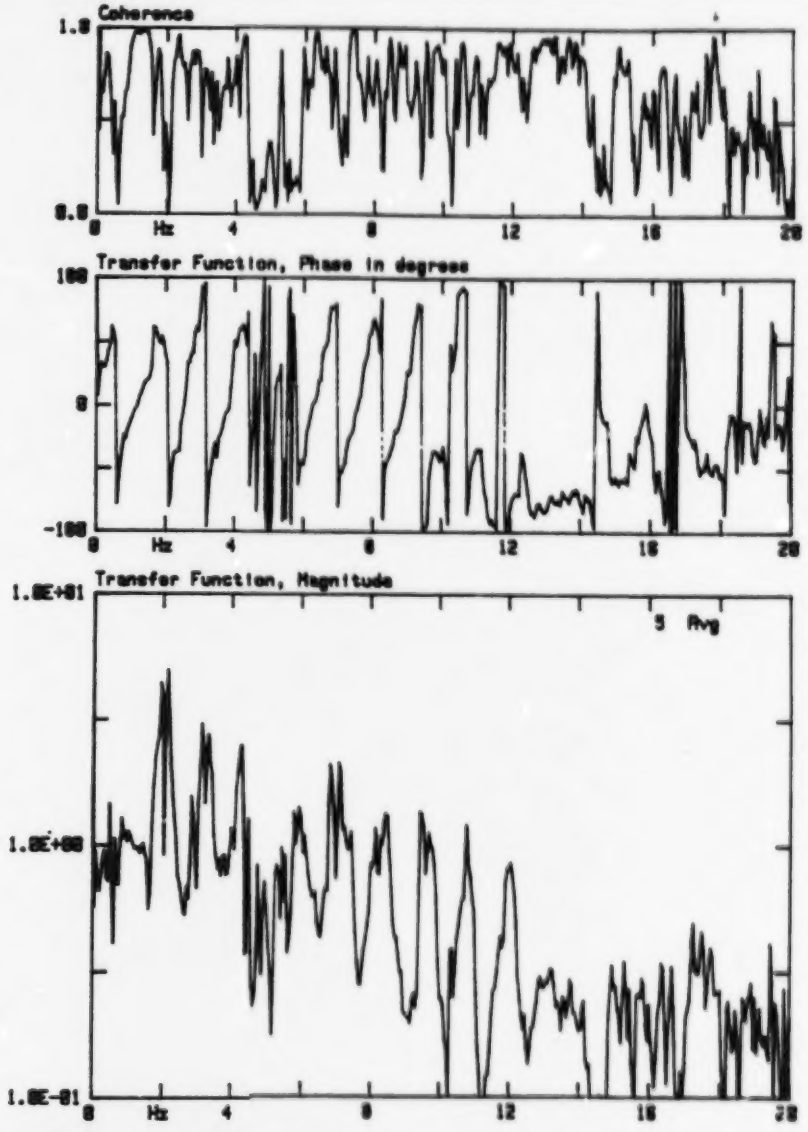
GOp0000d , GOk_03050Transfer Function between Intermediate & Upper Deck
GO Transit Testing 1988.06.16
Channel A is Channel 03, Vert. Acc. UD, Channel B is Channel 05, Vert. Acc. ID,
Starting time 203.10.13.45 to 14.45 about 600 revolutions out Port Credit
MTC (416) 235-3957 1988.09.22

Figure. 3-



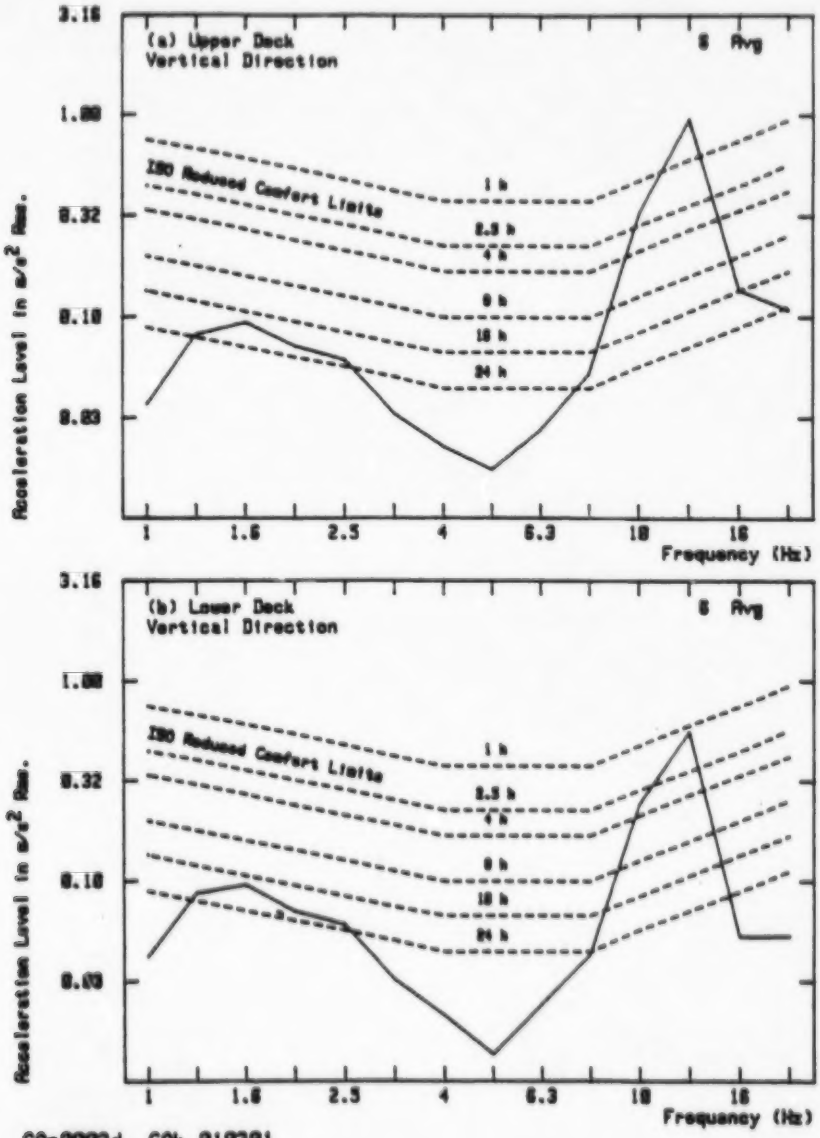
GOp0000d , GOk_010302, Transfer Function between Upper & Lower Deck
GO Transit testing 1988.06.21
Channel A is channel 01, Channel B is channel 03, Clarkson to Port Credit,
10:40:40 to 10:41:40, Baseline Data
MTC (416) 235-3557 1988.09.22

Figure. 27



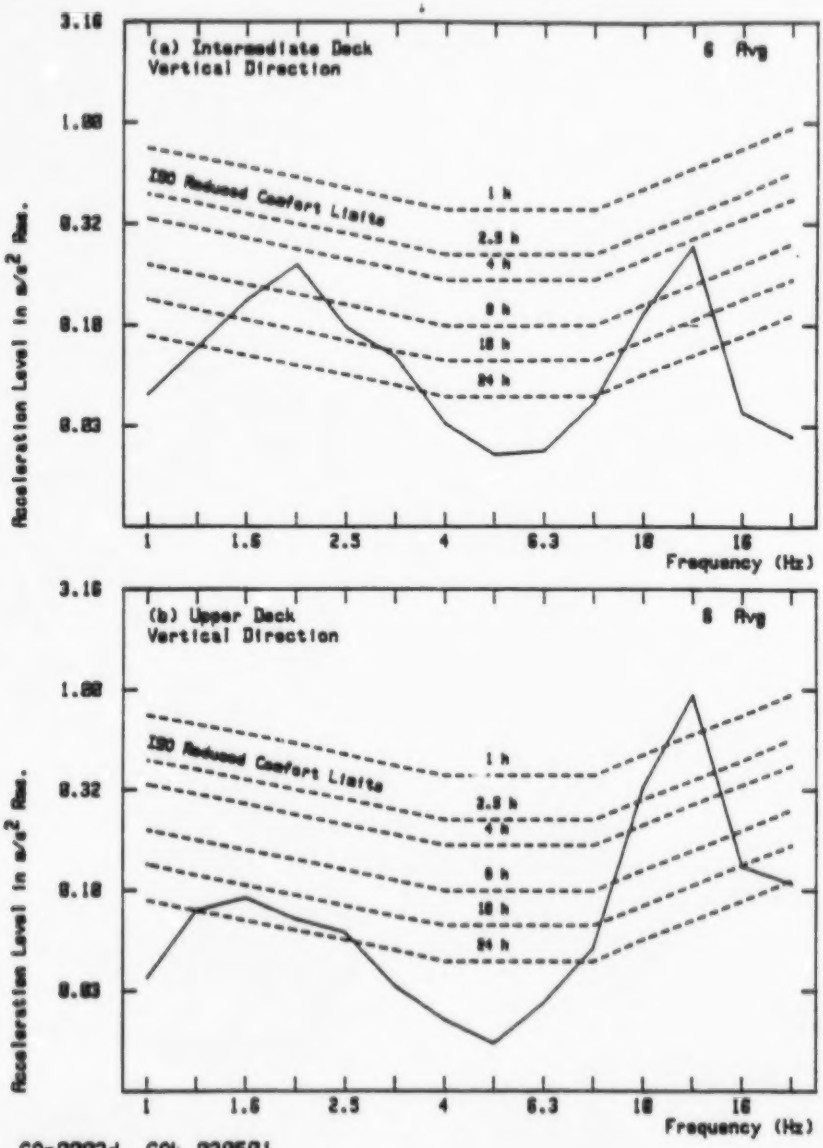
GOp8008d , GOk 83050 Transfer Function between Intermediate & Upper Deck
GO Transit testing 1988.06.21
Channel A is channel 3, Channel B is channel 5, Clarkson to Port Credit,
18:40:40 to 18:41:40
MTC (416) 235-3957 1988.09.22

Figure. 28



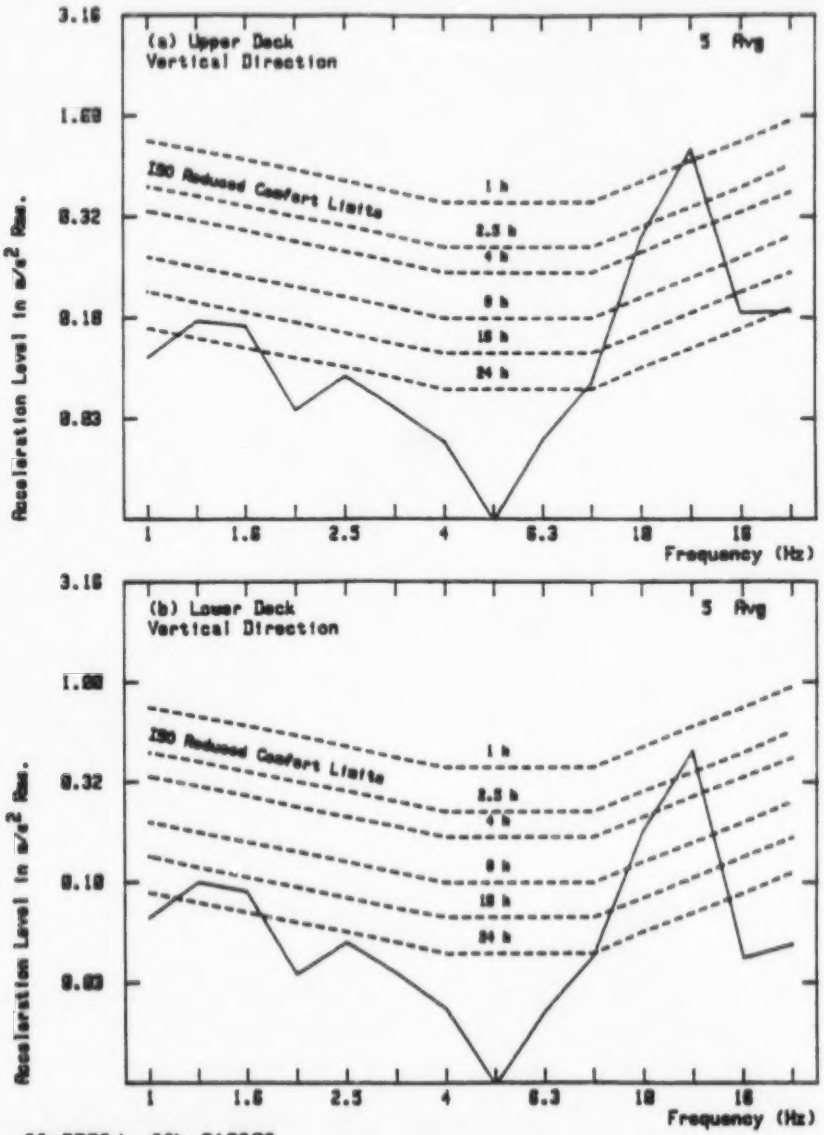
GOp0802d, GOk_810301
GO Transit Testing 1989.06.21
Channel A is Channel 01, Channel B is Channel 03, Port Credit to Clarkson
10:13:45 to 10:14:45, Baseline Data.
MTC (416) 235-3957 1989.06.22

Figure. 29



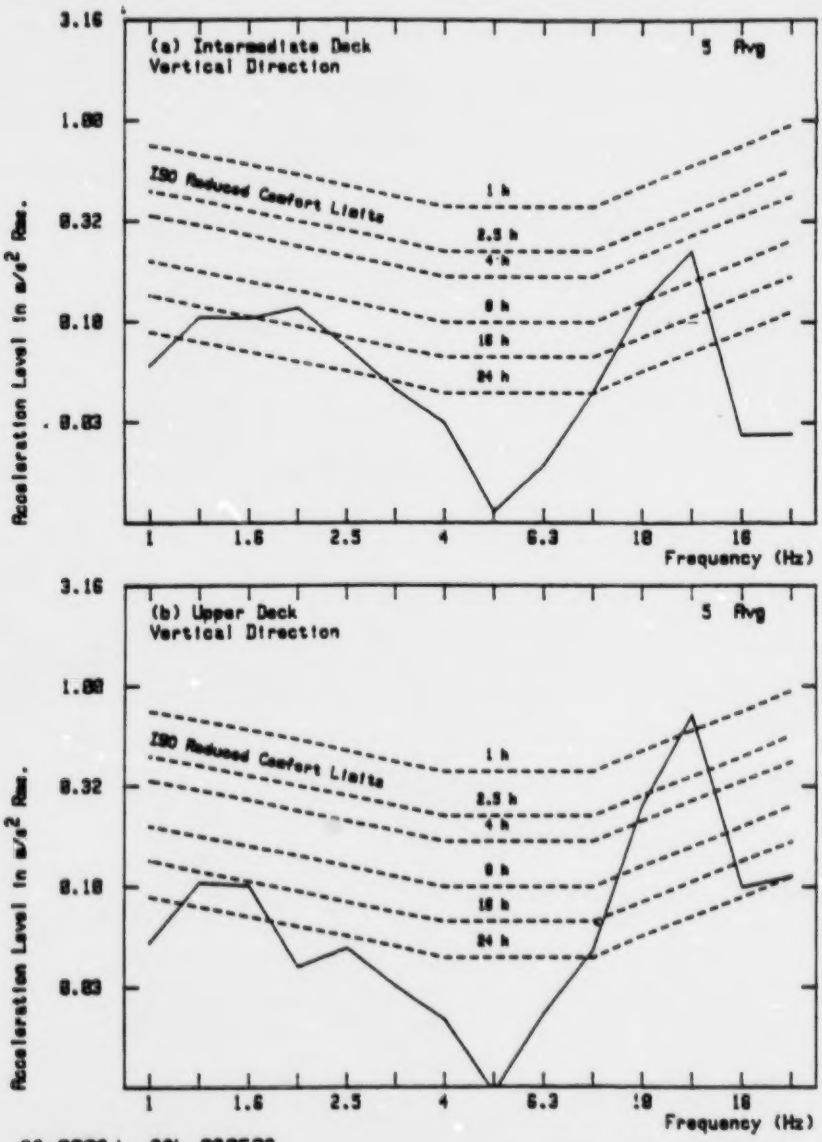
GOp0002d, GOk_830501
GO Transit Testing 1988.05.16
Channel A is Channel 03, Vert. Acc. UD, Channel B is Channel 05, Vert. Acc. ID,
Starting time 203.10.13.45 to 14.45 about 600 revolutions out Port Credit
MTC (416) 235-3957 1988.09.22

Figure. 30



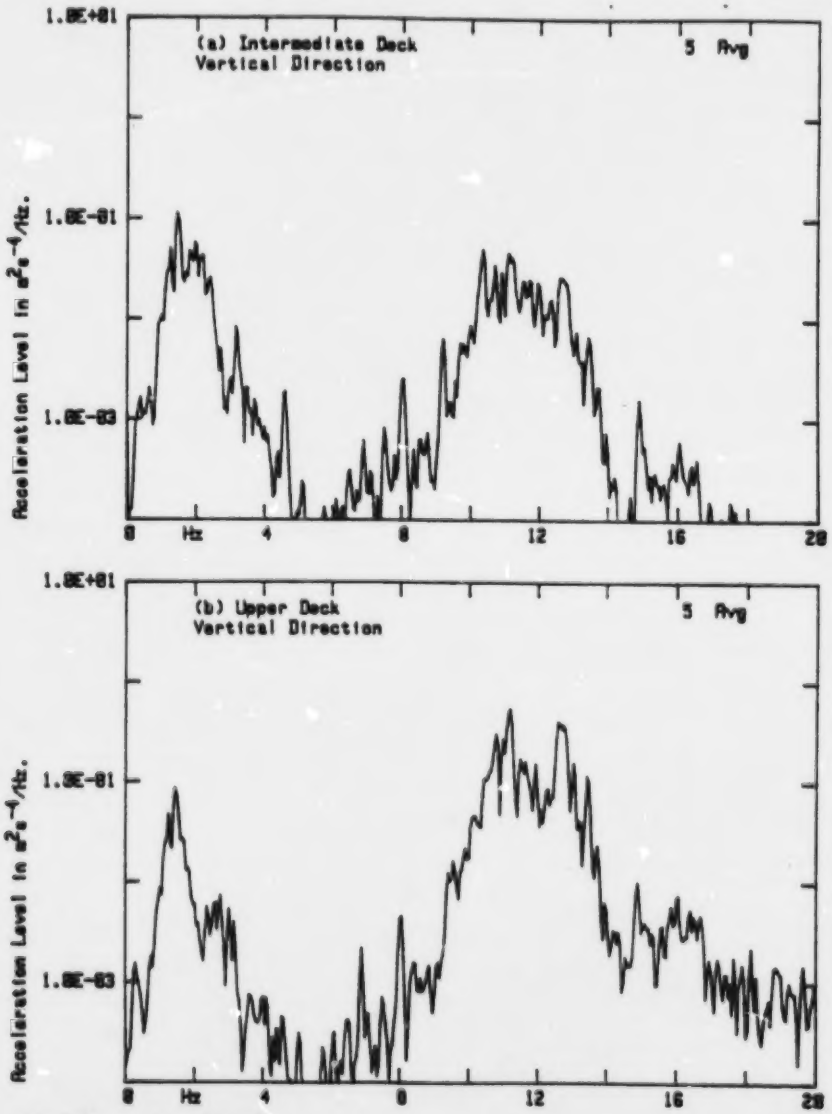
GOp0002d, GOk_010302
GO Transit testing 1988.06.21
Channel A is channel 01, Channel B is channel 03, Clarkson to Port Credit, 10:40:40 to 10:41:40, Baseline Data
MTC (416) 235-3957 1988.09.22

Figure. 31



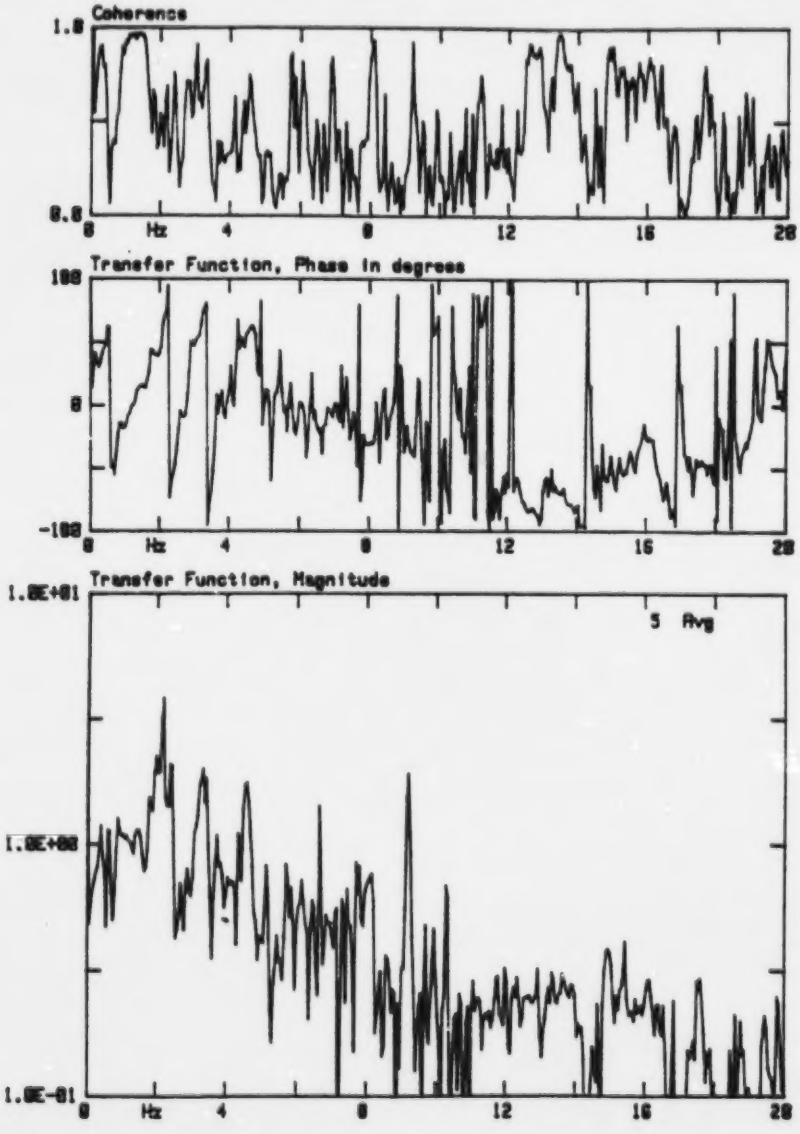
GOp0002d, GOk_030502
GO Transit testing 1988.06.21
Channel A is channel 3, Channel B is channel 5, Clarkson to Port Credit, 18:48:48 to 18:41:48
MTC (416) 235-3957 1988.09.22

Figure. 32



GOp0001d, GOk_030503
GO Transit testing 1988.06.22
Channel A is channel 3, Channel B is channel 5, Clarkson to Port Credit,
13:44:57 to 13:45:57 , Kent
MTC (416) 235-3957 1988.09.22

Figure. 33



GOp0000d , GOk_030503nsfer Function between Intermediate & Upper Deck
GO Transit testing 1988.06.22
Channel A is channel 3, Channel B is channel 5, Clarkson to Port Credit,
13:44:57 to 13:45:57 , Koni
MTC (416) 235-3957 1988.09.22

Figure. 34

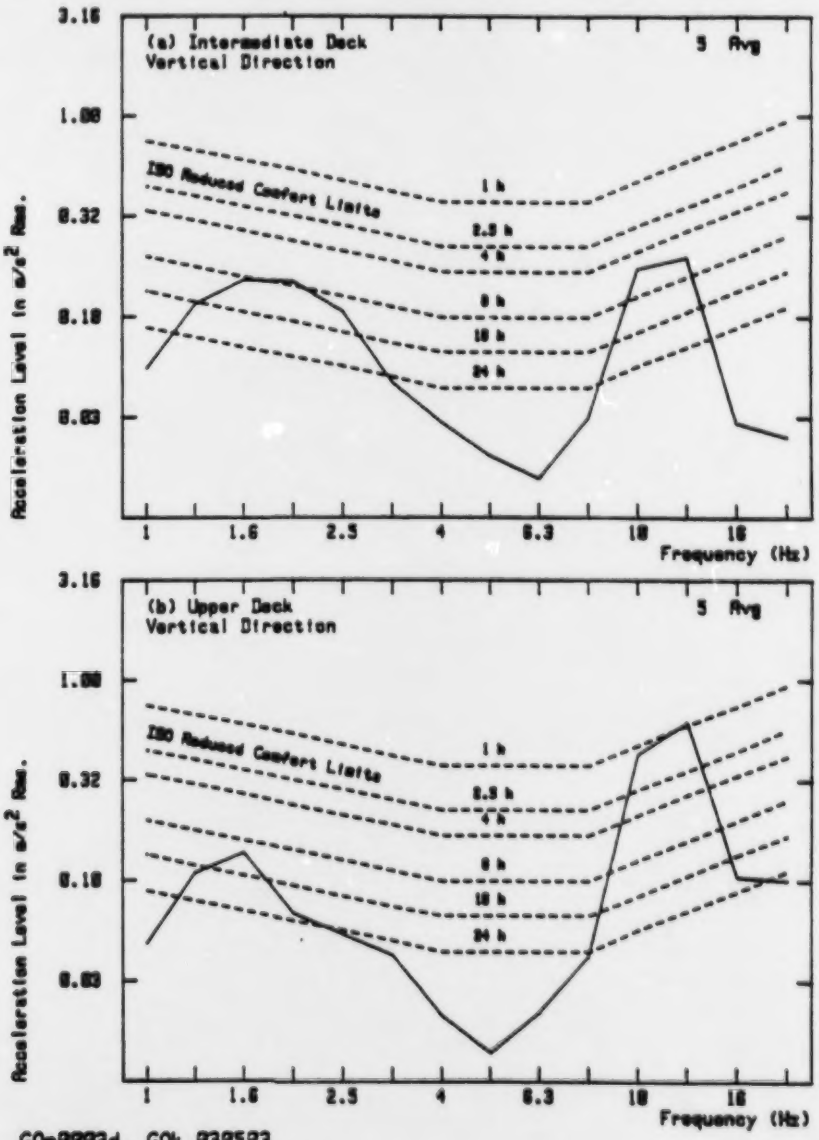
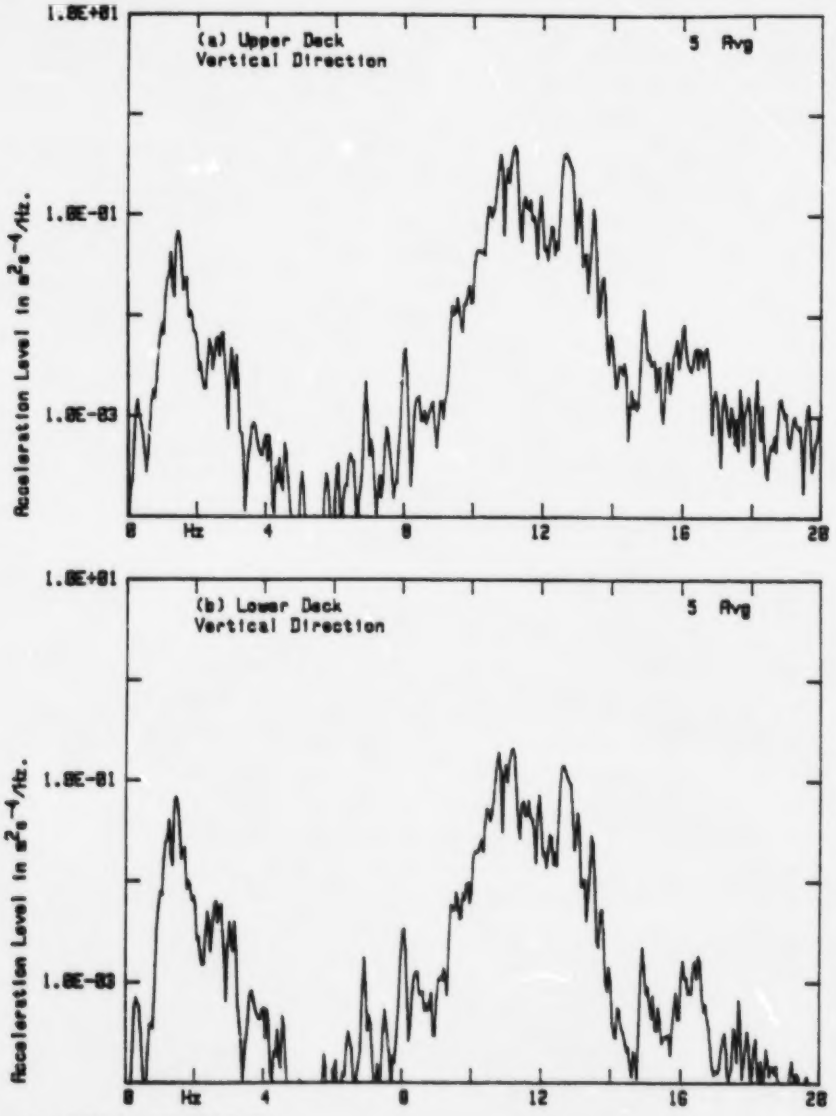
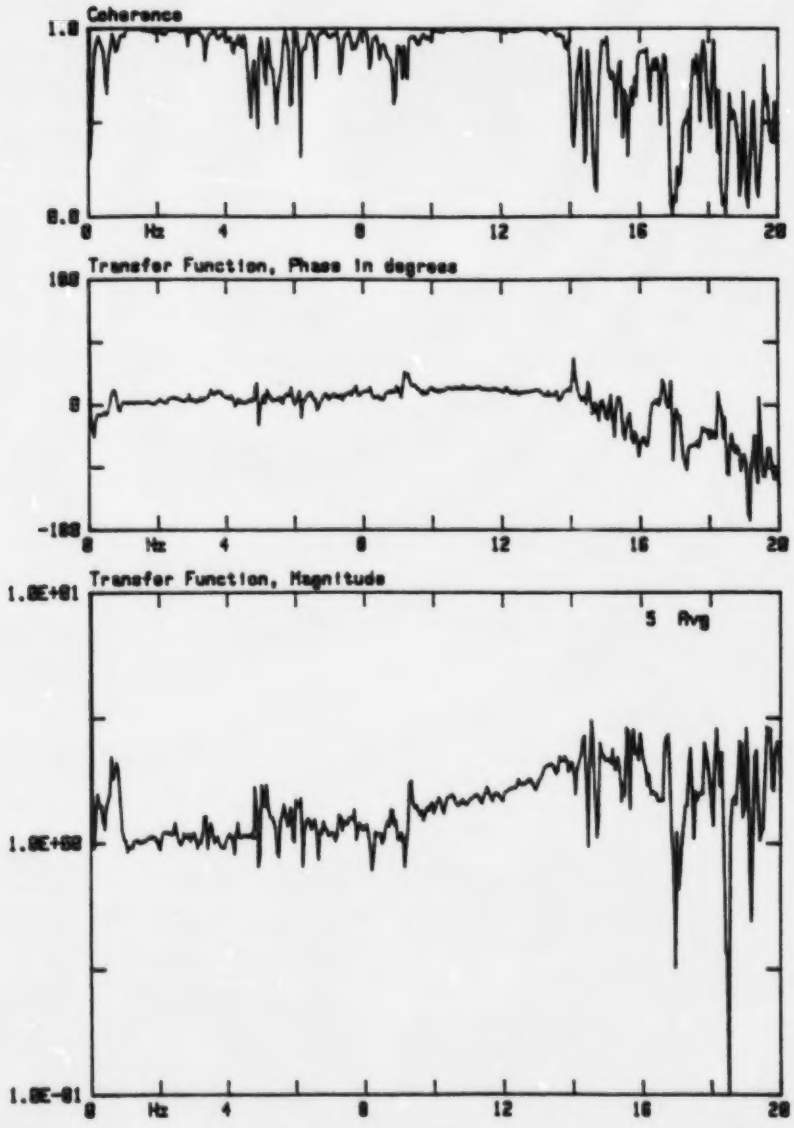

(a) Intermediate Deck
Vertical Direction
5 Avg
ISO Reduced Comfort Limits
1 h
2.5 h
4 h
8 h
16 h
24 h
Acceleration Level in m/s² Rms.
Frequency (Hz)
(b) Upper Deck
Vertical Direction
5 Avg
ISO Reduced Comfort Limits
1 h
2.5 h
4 h
8 h
16 h
24 h
Acceleration Level in m/s² Rms.
Frequency (Hz)
GOp0002d, GOk_030503
GO Transit testing 1988.08.22
Channel A is channel 3, Channel B is channel 5, Clarkson to Port Credit,
13:44:57 to 13:45:57 , Kent
MTC (416) 235-3957 1988.08.22

Figure. 35



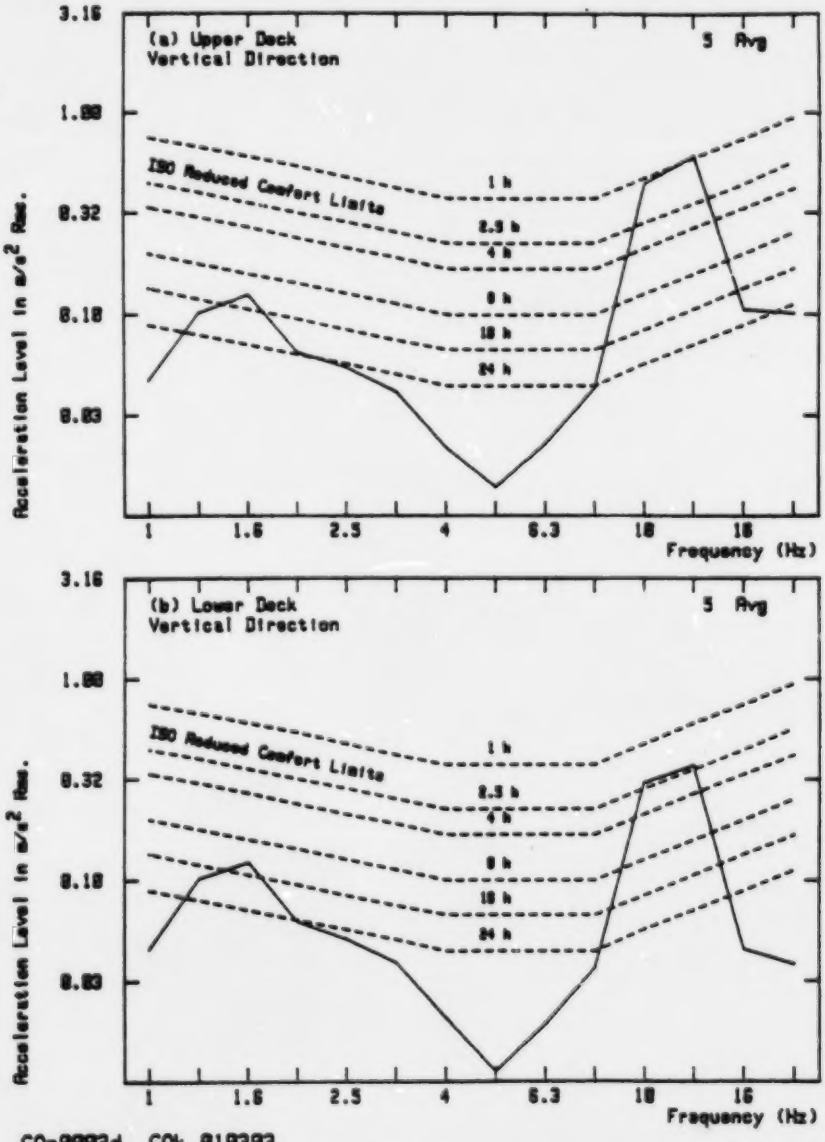
GOp0001d, GOk_010303
GO Transit testing 1988.06.22
Channel A is channel 01, Channel B is channel 03, Clarkson to Port Credit, 10:40:40 to 10:41:40 , Modified Kent Shocks -Stanchion
MTC (416) 235-3957 1988.09.22

Figure.



GOp0008d , GOk_010303, Transfer Function between Upper & Lower Deck
GO Transit testing 1988.08.22
Channel A is channel 01, Channel B is channel 03, Clarkson to Port Credit,
18:48:48 to 18:41:48 , Modified Koni Shocks - stanchions
MTC (416) 235-3957 1988.09.22

Figure. 37



GOp0802d, GOk_810303
GO Transit testing 1988.06.22
Channel A is channel 01, Channel B is channel 03, Clarkson to Port Credit, 10:40:40 to 10:41:40 , Modified Koni Shocks + Stanchions
MTC (416) 235-3957 1988.09.22

Figure. 34

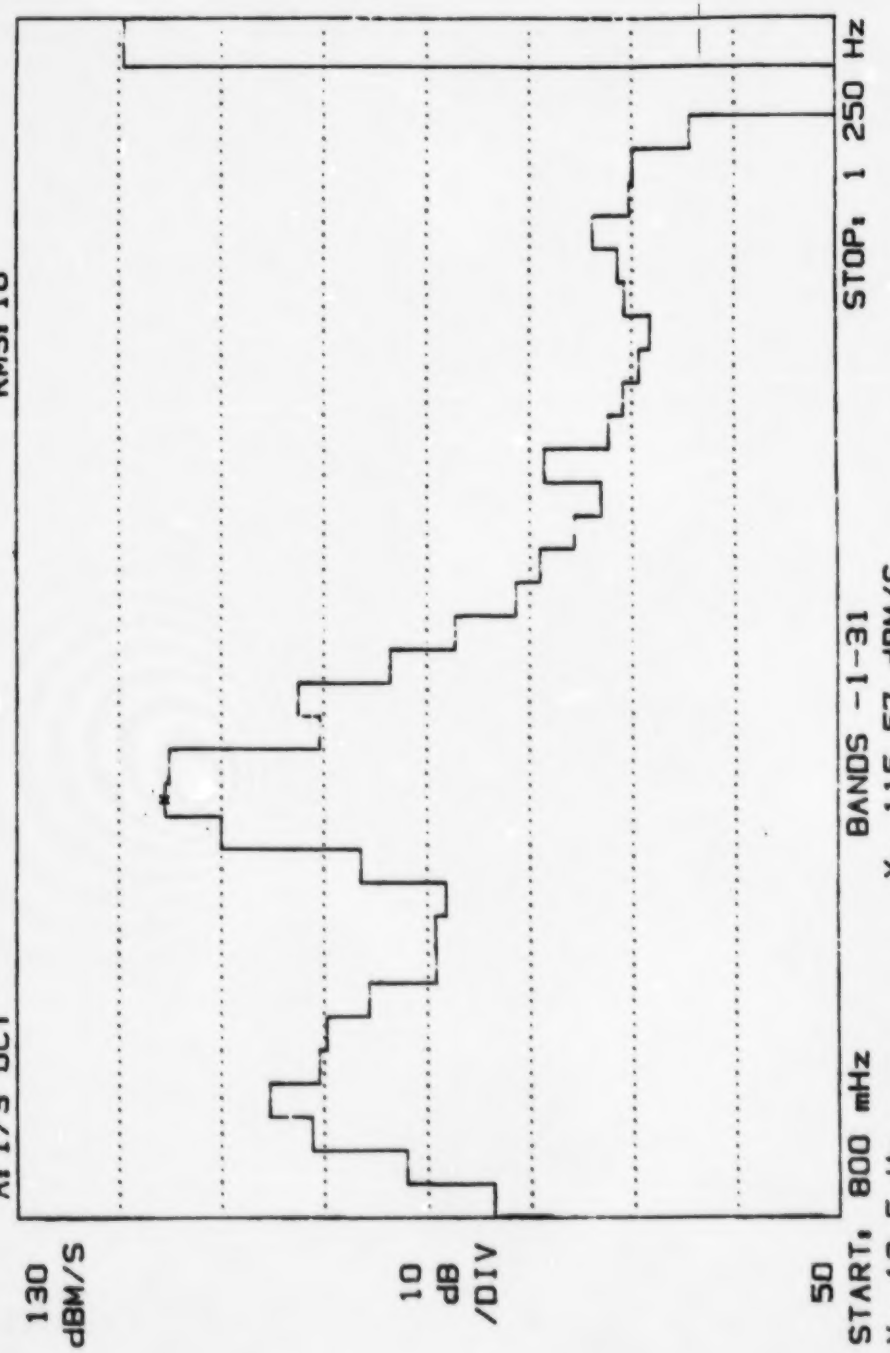


Figure. 39 Bilevel III Ride - Upper Deck #5 Vertical with Houdaille Mod

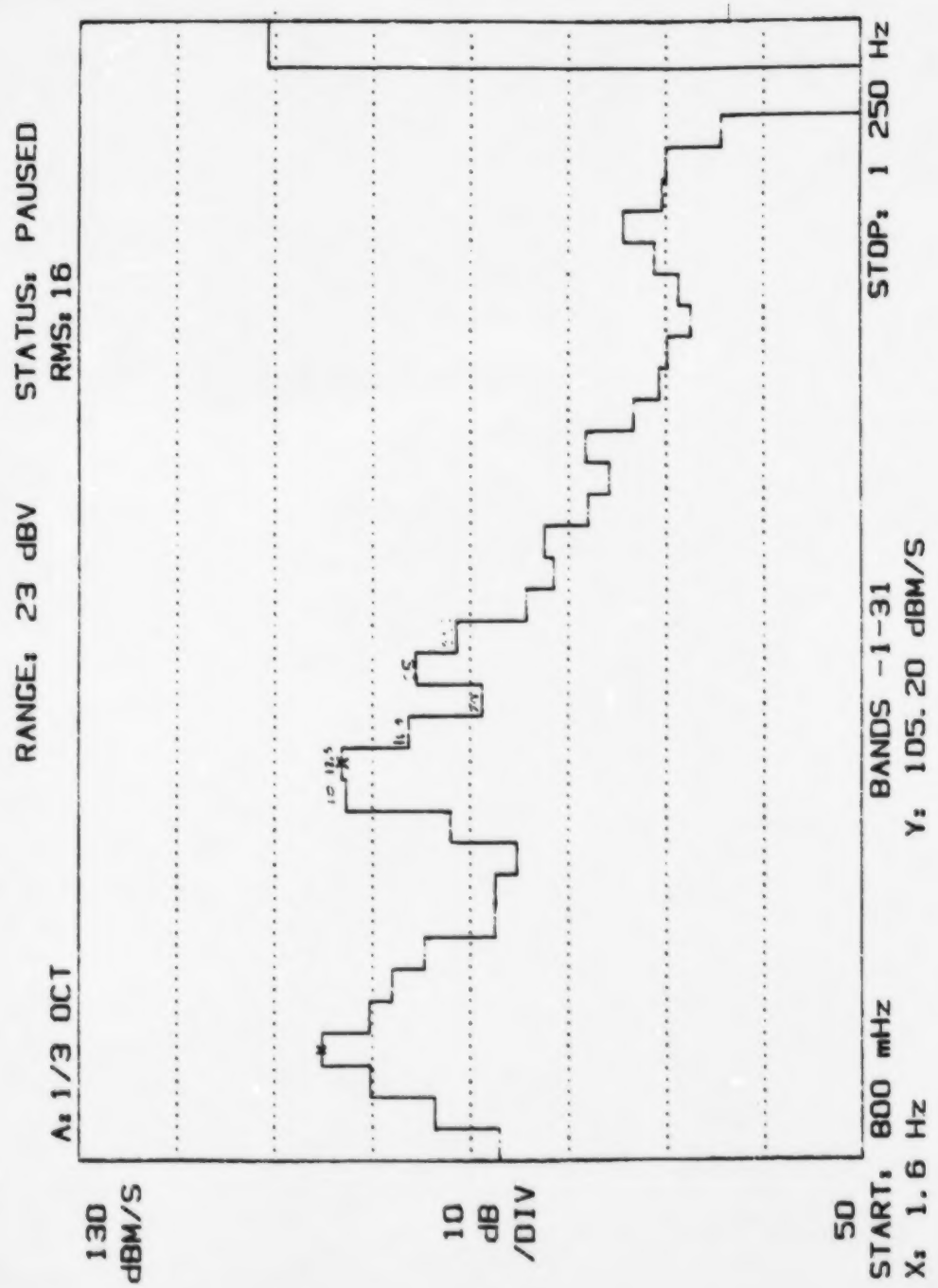


Figure. 40 Bilevel III Ride - Upper Deck Near Sidewall-
#7 Vertical with Houdaille Mod

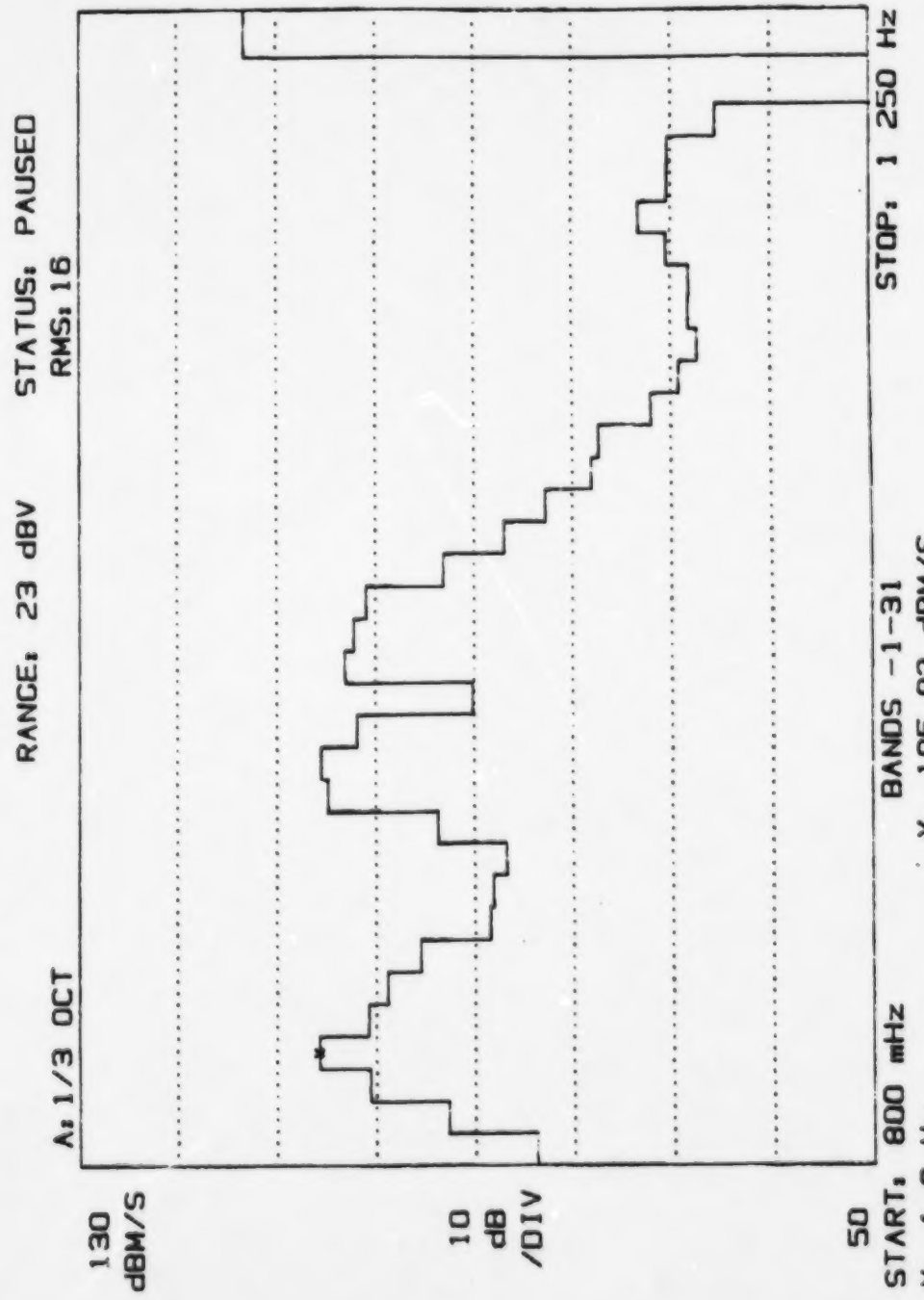


Figure. 41 Bilevel III Ride - Lower Deck #8 Vertical with Houdaille Mod

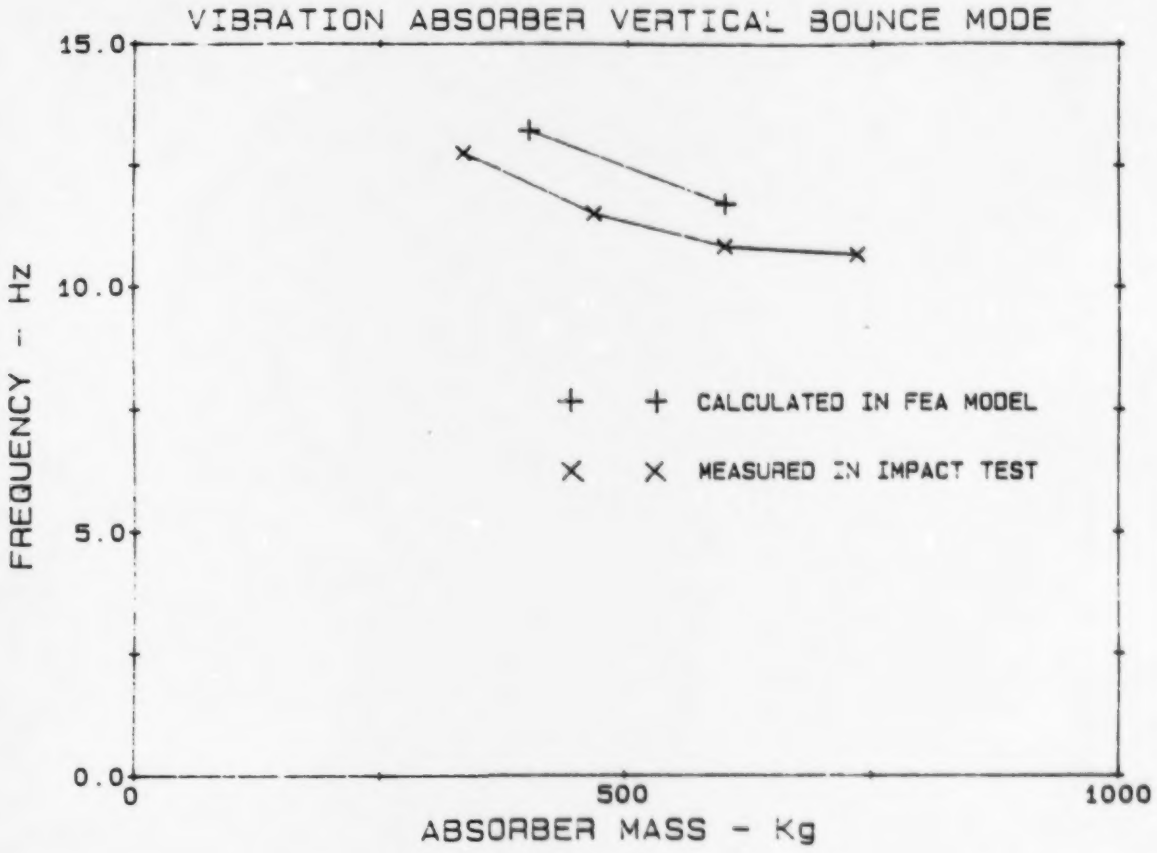


Figure. 42 Vibration Absorber Measured Versus Calculated Frequency

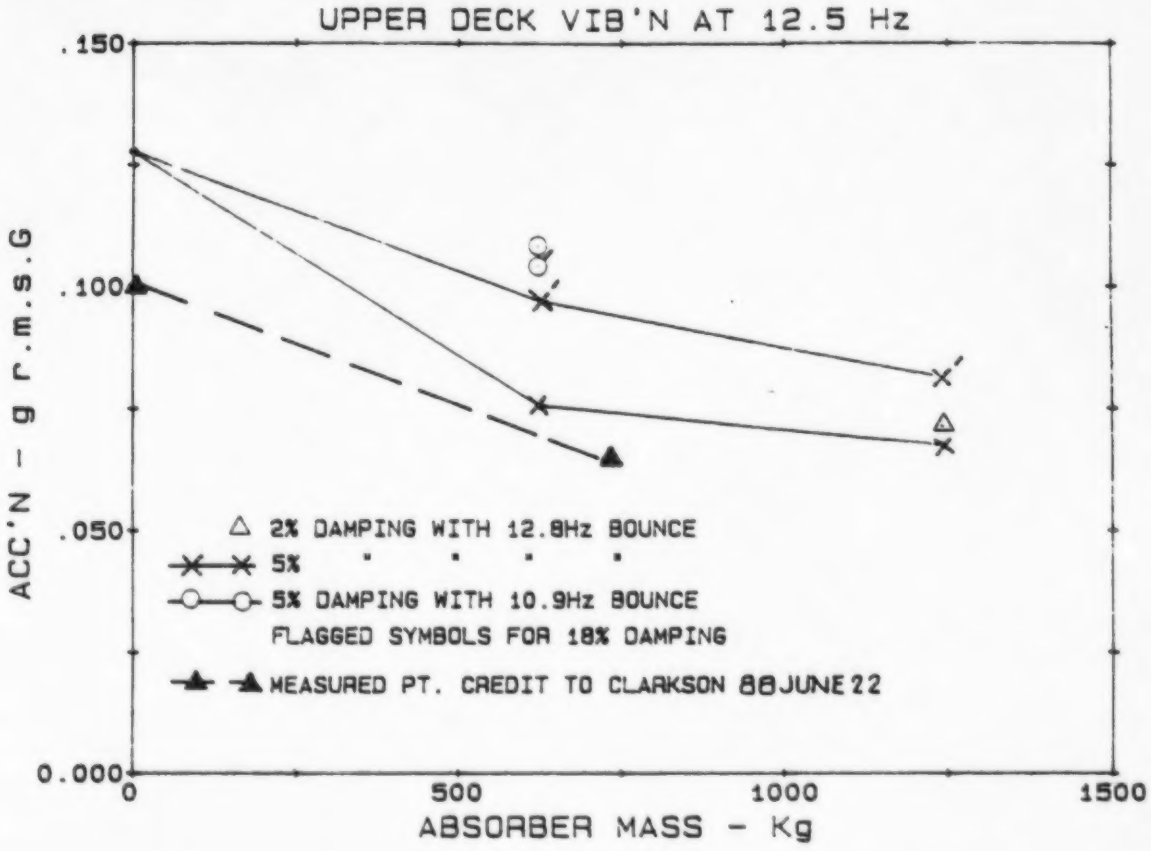


Figure. 43 Vibration Absorber: Measured Versus Calculated Ride

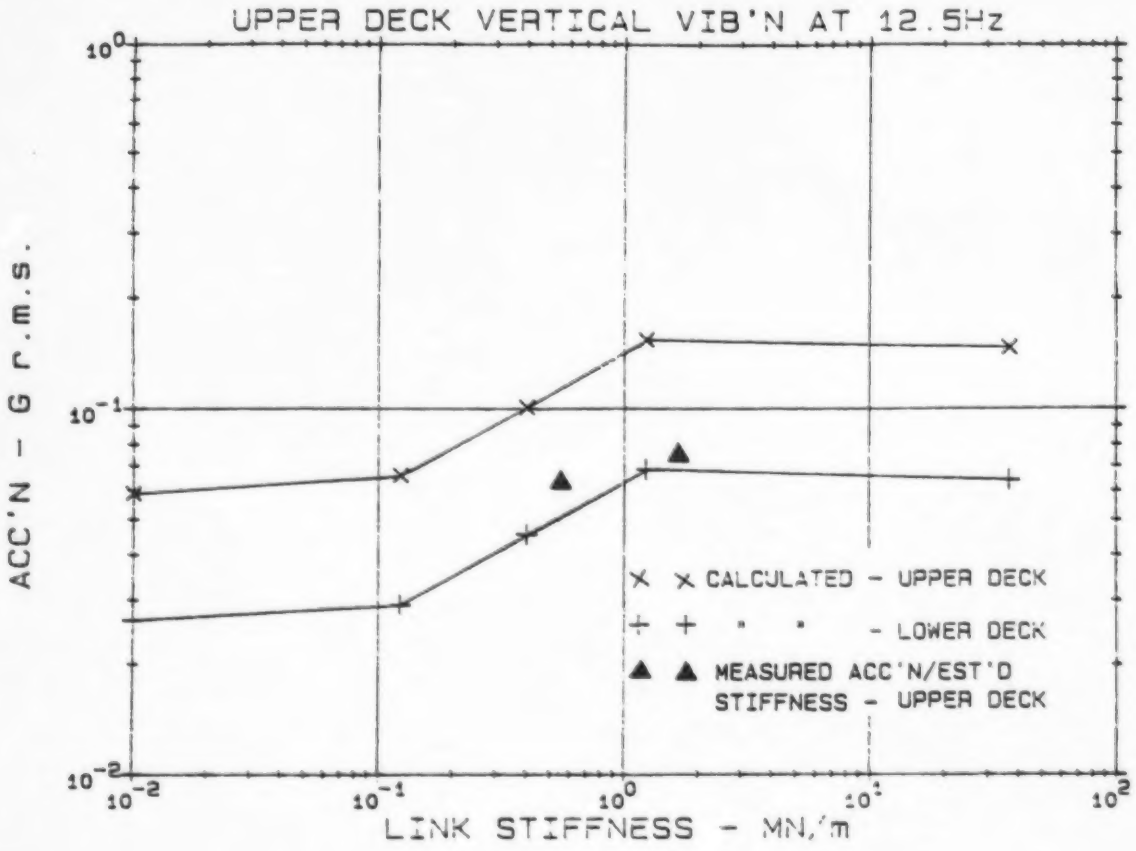


Figure. 44 Secondary Damper Link Mod: Measured Versus Predicted Ride